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MECHANICAL EQUIPMENT OF BUILDINGS

A REFERENCE BOOK FOR ENGINEERS AND ARCHITECTS

BY

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VOLUME I

HEATING AND VENTILATION

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PREFACE TO VOLUME I

This book is a new departure in the literature on the mechanical equipment of buildings. It proposes to deal not only with the heating and ventilation of buildings, which are considered in this first volume, but also (in subsequent volumes) with power plants, elevators, lighting systems, refrigeration plants, sprinkler systems, vacuum cleaning, and plumbing.

The object of the authors is to produce a reference book for engineers and architects which will contain sufficient theoretical and commercial data for practical use in the designing room, and at the same time serve to show the student of this subject the relation between the theoretical principles involved and their practical application to actual problems.

All available sources of information relating to this field of engineering have been drawn upon, and credit given in the text, wherever such information is introduced. The authors have found it necessary, in their own experience, to make extensive use of manufacturers' data in designing the various mechanical systems or plants required in modern buildings. They have therefore not hesitated to include such data in the text in order to illustrate and facilitate the design of similar systems in each subject treated.

References to specific makes of such equipment have not been intended as in any sense exclusive of other equipment of the same sort, but merely as indicating that the equipment named and described is as satisfactory as any to be obtained in the market.

The authors are especially indebted to Prof. G. A. Goodenough for many valuable suggestions, as well as permission to make use of his latest tables of the properties of steam and ammonia and also of air and vapor mixtures.

The Authors.

Urbana, Ill.,
July, 1916.

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HEATING AND VENTILATION

CHAPTER I.

PHYSICAL UNITS AND THE MEASUREMENT OF HEAT

FUNDAMENTAL UNITS

MODERN engineering practice depends on the correct application of basic principles already developed and established in such branches of physical science as mechanics, thermodynamics, hydraulics, physics and chemistry. As reference will be made to these fundamental principles from time to time, it is necessary to define the units in which the various quantities dealt with will be measured.

In this country the system of units in general use by engineers is known as the *Foot-Pound-Second System*, and the following definitions and examples will show the significance of each unit. A table of equivalents (Table 2) is also given, so that the value of the more general compound units can be found in terms of various other units.

Definitions of Units and Terms Employed in the F. P. S. System. The unit of time is the second, which is equal to $\frac{1}{86,400}$ part of the mean solar day. t = *time*. Time is also expressed in minutes and hours.

L = *length*. The unit of length is the foot = 0.3048 meter.

W = *weight*. The unit of weight is the pound = 0.4536 kilogram.

A = *area*. The unit of area is the square foot. The unit often used is the square inch.

V = *volume*. The unit of volume is the cubic foot. Volume equals area × length = $A \times L$. In calculations involving quantity of air required Q is often used for cu. ft.

Example. The volume displaced per stroke by the plunger of a pump, if the diameter is 6″ and the stroke is 12″, is $\frac{1}{4}\pi \times 6^2 \times 12 = 339.29$ cubic inches or 0.196 cu. ft.

If the plunger makes 30 "working" strokes (not revolutions) per minute, then the plunger "displacement" per minute is $0.196 \times 30 = 5.88$ cu. ft. One U. S. gallon = 231 cu. inches or 0.1336 cu. ft. This pump will therefore theoretically deliver $\frac{5.88}{0.1336}$ or 44 gals. per min. The actual delivery of the pump will be somewhat less owing to "slip," which is the leakage back through the pump valves, around the plunger, and that due to imperfect filling of the pump cylinder on the suction stroke.

D = *density*. The weight of a unit volume (1 cu. ft.) of a substance is called its "density." The density of water at 70° F. is 62.3 lb. per cu. ft., and at 60° F. = 62.37.

The densities of the following liquids at 60° F. are:

Petroleum: 48.7 to 54.9 lb. per cu. ft.

Mercury: 848.7 lb. per cu. ft. Specific gravity = 848.7/62.37 = 13.6.

The pump in the preceding example would, therefore, handle 5.88 × 62.3 or 366 lb. of water per minute.

If the water end of the pump were operated by a steam cylinder having a displacement of 0.349 cu. ft. per stroke and took steam at the same pressure for the full stroke as in the "direct acting" type and assuming that the steam pressure were 100 lb. gage, we find from the steam tables that the density of steam at this pressure is 0.2565 lb. The "steam consumption" of the pump, therefore, would be 0.2565 × 0.349 × 30 × 60 or 161.6 lb. per hr. theoretically.

v = *velocity.* The rate of motion of a body is measured by the distance passed over in a unit time. Velocity is expressed in ft. per sec.

a = *acceleration.* The rate of change of velocity measured in ft. per sec. is termed acceleration, and is stated in ft. per sec. per sec. (generally expressed, ft. per sec.^{2}). Acceleration may be either positive or negative, depending upon whether the speed of the moving body is increasing or decreasing. The uniform acceleration due to gravity, denoted by the symbol g, is the rate of gain in velocity of a freely falling body and is 32.174 ft. per sec.^{2} The value of g is generally taken as 32.2.

M = *mass.* The expression W/g is termed "mass." A unit of mass is the quantity of matter in pounds to which the unit of force (1 lb.) will give an acceleration of 1 ft. per sec.^{2}

Relation between Velocity, Acceleration, Time, and Space Passed Over. When the accelerating force is uniform the acceleration will be uniform. The velocity at the end of t seconds, if the body starts from rest, will be

$$v = at; \text{ whence } a = \frac{v}{t}, \text{ and } t = \frac{v}{a}$$

The space passed over at the end of t seconds is equal to the product of the mean velocity and the time, or $L = \frac{1}{2}vt$, or $L = \frac{1}{2}at^2$.

A force of 1 pound when applied to a mass whose weight is 32.17 pounds will produce an acceleration of 1 ft. per sec.^{2} when the mass is moving against no resistance (frictionless motion). A force of F pounds acting on a mass of one pound will produce an acceleration of $F \times 32.17$ ft. per sec.^{2}

The relation between force, mass, and acceleration is given by the equation $F = Ma = Wa/g$. Substituting the value of a in terms of v we obtain

$$F = \frac{W\,v}{g\,t}$$

U = *energy* or *work.* The unit of work is the foot pound, and is the quantity of energy expended or the work performed by a force of 1 pound moving through a distance of 1 foot in the line of action of the force.

Power is the rate of doing work. Note that "power" involves the factor "time" and is equal to the amount of work done divided by the time required to do this work.

h.p. = *Horsepower.* The unit of power is the "horsepower" and is the performance of work at the rate of 550 ft.-lb. per sec. or 33,000 ft.-lb. per minute.

Example. Required the theoretical work and horsepower developed by the water end of the pump in the preceding example if the head or the height pumped against is 200 ft., assuming no frictional resistance to be overcome.

The work U_m performed per minute is the lifting of the weight of water, W = 366 lb. per minute, through a height of 200 ft. is

$$U_m = 366 \times 200 = 73{,}200 \text{ ft. lb. per min. and h.p.} = \frac{U_m}{33{,}000} = \frac{73{,}200}{33{,}000} = 2.22.$$

The actual power required will be somewhat greater, as we have neglected the force required to overcome frictional resistance, and the force required to accelerate the water from a state of rest to the velocity at which it is delivered.

The total force F required on the plunger will be the total pressure produced on the plunger by the water column, neglecting friction and the accelerating force.

The pressure per unit area produced by the water column is equal to the height H in ft. multiplied by the density D of the water or $P = H\,D = 200 \times 62.3 = 12{,}460$ lb. per sq. ft., or $p = \dfrac{12{,}460}{144} = 86.5$ lb. per sq. in.

If A = area of plunger in sq. in., then $F = p\,A$ or $86.5 \times 28.27 = 2{,}446$ lb.

The work performed per stroke of the plunger is $U = F\,L$ in which L is the length of the stroke in ft. or $U = 2{,}446 \times 1 = 2{,}446$ ft.-lb.

The work per min. $= U_m = 2{,}446 \times 30 = 73{,}380$ and the power required is $\dfrac{73{,}380}{33{,}000} = 2.22$ h.p.

Measurement of Pressure. It is customary to measure pressure by means of gages which in reality only indicate the difference between the pressure being measured and the pressure of the atmosphere (barometric pressure) at the same time and place. These gages may indicate either a higher or lower pressure than that of the atmosphere; in the former case they are known as *pressure gages* and in the latter as *vacuum* or *draft gages*.

Pressure and Vacuum Gages. The most common type of pressure gage (Fig. 1) is provided with a flexible hollow brass tube of oval cross section known as a *Bourdon* tube. When subjected to pressure, this tube tends to straighten out and thus causes a sector of a gear to mesh with a small pinion on the same shaft with the indicating hand or pointer and rotate the latter a corresponding amount. The pointer is placed just in front of a graduated dial (not shown in the figure) from which the pressure may be read in suitable pressure units such as pounds per square inch.

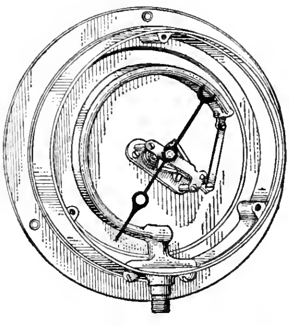


FIG. 1. SINGLE SPRING PRESSURE GAGE. INTERIOR VIEW.

These gages may also be used for indicating vacuum or pressures less than that of the atmosphere.

Draft Gages. The measurement of pressures but slightly above or below the atmospheric pressure (barometric pressure) is usually accomplished by the use of a draft *gage* (Fig. 2) connected at the stop cock on the right hand side.

This is essentially a U tube, containing either water, kerosene, alcohol or mercury, mounted upon a graduated scale, and reading either in inches of fluid or in pounds or ounces per square inch. Since the pressure indicated is a differential one, due to the left hand leg being open to the air, the reading must be obtained by adding the depression in the left hand leg to the elevation in the right hand leg; using zero as the reference point in both cases. Thus, for the gage shown in Fig. 2 the reading is $0.95 + 1.05 = 2.00$ inches of mercury from which the vacuum or pressure below the atmosphere in pounds or ounces per square inch may be readily calculated.

Barometers. The pressure of the atmosphere is usually measured by a *mercurial barometer* (Fig. 3) which in its simplest form consists of a glass tube about 3 feet long, closed at one end, which after being filled with mercury is inverted in a shallow bath of mercury. The pressure of the atmosphere at sea level maintains the mercury column in the tube about 30″ above the level in the cistern. The barometric height or length of this column of mercury varies with the altitude above or below sea level.

When the mercury in the tube falls, that in the cistern rises in corresponding proportion, and vice versa, so that there is an ever-varying relation between the level of the mercury in the tube and the mercury in the cistern, which affects the accuracy of the readings. It is therefore necessary before reading the height of the mercury column on the stem of the barometer (Fig. 4) by means of the movable vernier C to adjust the level of the mercury in the cistern.

The cistern (Fig. 5) consists primarily of a heavy walled glass cylinder AA allowing the surface of the mercury B to be clearly seen.

This cylinder is securely held between bolsters CC in a movable frame suitably mounted in the base of the instrument.

The bottom of the frame D is connected with the threaded stem E which ends in the knurled

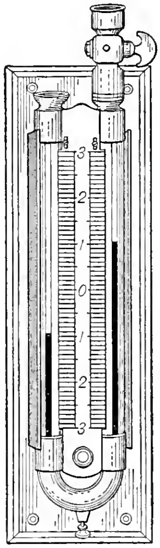


FIG. 2. DRAFT GAGE.

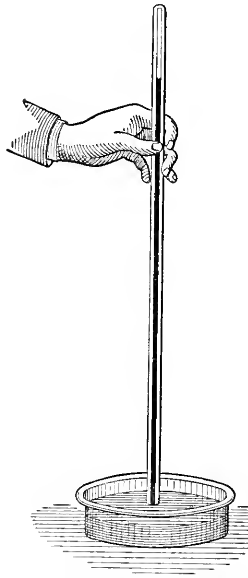


FIG. 3. SIMPLE BAROMETER.

nut F, by means of which the cistern is moved vertically thus raising or lowering the mercury level and adjusting same to the tip of the ivory pointer G, which is the zero of the scale.

All standard or observatory barometers of the mercurial type possess this adjustable feature. Barometers of other types, such as the *Aneroid barometer,* must be frequently compared with a standard mercurial barometer in order to check the accuracy of their readings.

Barometric Pressure. By barometric height is meant the height of a column of pure mercury at 32° F. which just balances the pressure of the atmosphere at the time and place of the observation. The *standard* or *normal barometric pressure* is defined as the pressure of a column of pure mercury 760 mm. (29.92 inches) high at 32° F. This is the normal barometric pressure at latitude 45° and sea level. Since the weight of 1 cu. in. of mercury under these same conditions is 0.491 lb. then the normal barometric pressure equals the height of mercury column $\times$ weight per cubic inch, = 29.92 $\times$ 0.491, or 14.7 lb. per sq. in.

This pressure of 14.7 lb. per sq. in. is known as the *absolute pressure* of the atmosphere at latitude 45° and sea level. Now, since the ordinary pressure gage measures only pressures above or below that of the atmosphere it is necessary to *add the barometric pressure* at the place in question *to the gage reading to obtain the total absolute pressure* corresponding to the pressure indicated by the gage. That is: absolute pressure = barometric pressure + gage pressure.

The pressures used must be in the same units, and may be expressed in pounds per sq. ft., P or *specific pressure,* or in pounds per sq. in., p, the usual units for expressing *gage pressure* $P = 144\,p$. Also pressure in inches of mercury $\times$ 0.491 = pressure in pounds per sq. in.

HEAT

Definition of Heat. *Heat is a form of energy,* and not a substance. It is, in fact, the kinetic and potential energy of the molecules of which all substances, whether solid, liquid, or gaseous, are composed. Whenever the vibratory motion of the molecules composing a body of given mass is increased from any cause the *thermaê kinetic energy* is increased. The temperature of the body rises, its *sensible heat* increases, and the body feels warmer.

The *thermal potential energy* of a body of given mass may be increased by causing it to expand or change its state, thus separating the molecules against their mutual attractions and requiring the expenditure of work or its equivalent in heat. The work expended in separating the molecules due to expansion, or in changing their state of aggregation, as in changing from solid to liquid, is stored in the body as potential energy. There is no change in temperature during changes of state, hence the kinetic energy and the temperature remain constant.

Furthermore, the thermal kinetic energy of a body for a given rate of vibration of the molecules will vary with their number or the mass of the body. Hence if the rate of this molecular vibration is the same in two different masses of the same substance, they will have the same *heat intensity* or *temperature*, but the larger mass will have the greater *heat content* or possess more *heat energy*.

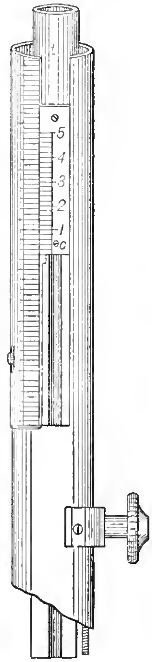


FIG. 4. STEM AND VERNIER.

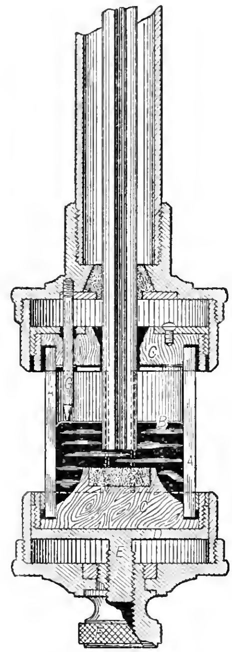


FIG. 5. CISTERN CONSTRUCTION.

OBSERVATORY BAROMETER.

Measurement of Temperature. (Thermometry.) *Intensity* of heat is measured by *thermometers* and *pyrometers,* the latter being used for high temperatures, above 400° to 500° F. In engineering work mercurial thermometers are very largely employed. These depend upon the uniform expansion of mercury to indicate changes in temperature. The unit of measurement is called a *degree,* and is capable of very exact determination, provided two points, at which the heat intensity is always constant, can be used as a base or reference for calibration. The melting-point of ice and boiling-point of water at atmospheric pressure are usually selected as bases, and the uniform expansion of the mercury between these two points is indicated on a scale divided into 180, 100, or 80 divisions. (Fig. 6.) Each of these divisions is known as a degree and the scales used are known respectively as *Fahrenheit, Centigrade* or *Celsius,* and *Réaumur.* The former is used almost exclusively in engineering work in this country.

Due to variations, under service, in the glass of which mercurial thermometers are made, it is necessary to compare them from time to time with a standard or to check the melting and boiling-point readings for accuracy. This *calibration* of a thermometer, as the process is called, should always be made before taking any important temperature readings with the instruments. The corrections are either tabulated or plotted, and the sign (+) or (−) prefixed, the former indicating that the correction is to be added, and the latter that it is to be subtracted from the observed reading. For example, if the thermometer actually reads 200°, and the correction table shows + 2.3 at this observed temperature, the actual temperature is 202.3°.

A further correction for stem exposure must also be made in very exact work, due to the fact that thermometer scales are graduated to read correctly for total immersion, that is, with bulb and stem of the thermometer at the same temperature, and they should be used in this

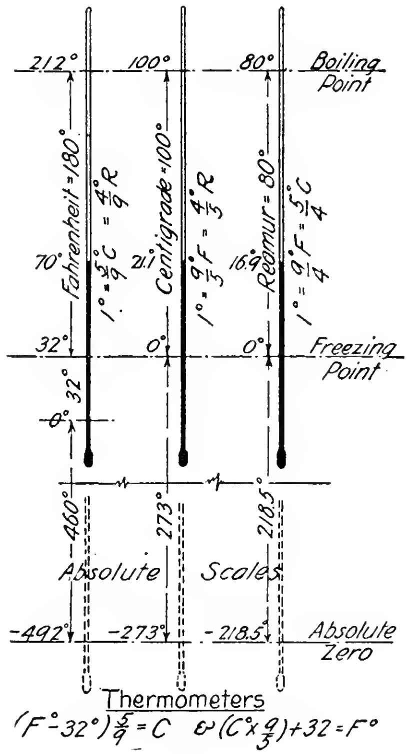


FIG. 6.

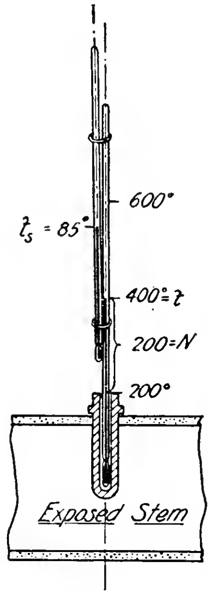


FIG. 7.

way when compared with a standard thermometer. If the stem emerges into space either hotter or colder than that in which the bulb is placed a "stem correction" must be applied to the observed temperature, and is made by use of the following formula:

$$\text{correction} = 0.000085\, N\, (t - t_s),$$

where the decimal is the difference between the coefficient of expansion of the mercury and the glass in the stem,

N = number of degrees of emergent mercury column,

t = observed temperature, and t_s = mean temperature of the emergent column. (Fig. 7.)

Absolute Temperature. In addition to the three temperature scales already described physicists employ what is known as the "absolute scale of temperatures," based on the so-called "absolute zero of temperature," at which point no molecular vibration exists. This zero is conceived as 491.6° F. below the melting-point of ice, or 32° F., it having been discovered that an ideal perfect gas would change in volume by $\frac{1}{491.6}$ of its volume at 32° for each 1° change in its

temperature at constant pressure. Thus, if 491.6 cu. ft. of gas measured at 32° F. is cooled 20° F. at constant pressure the new volume will be 471.6 cu. ft.

It is only necessary to add 491.6 − 32 or 459.6 to the actual thermometer reading to get the absolute temperature, that is, $T = t + 459.6$, where T = absolute temperature, and t = actual thermometer reading on the Fahrenheit scale. For engineering work 460° is used rather than 459.6°. For the Centigrade scale the relation is $T = t + 273.1$.

Pyrometers. For the measurement of high temperatures above 500° F. *pyrometers* of various kinds are employed. Mercurial pyrometers may be used for flue-gas temperatures up to 1000° F. These are simply thermometers with *an inert gas* such as nitrogen or carbon dioxide forced in above the mercury column to prevent the mercury from boiling, since at atmospheric pressure it will boil at 676° F. In fact, vaporization begins much below this temperature, so that ordinary thermometers should not be used much above 400° F.

Expansion pyrometers made up of two dissimilar metals, such as brass and iron, are used for temperatures up to 1500° F. They are liable to error unless both the brass and iron elements are uniformly heated throughout. In the common form a brass rod is enclosed in an iron pipe and one end of the rod attached to a cap at the end of the pipe, while the other end is connected by a multiplying gear to a pointer moving around a graduated dial. Lost motion in the gearing is often a source of error.

Thermo-electric pyrometers are used for temperatures up to 2900° F., and are described in "Steam," *Babcock & Wilcox Co.*, as follows:

"When wires of two different metals are joined at one end and heated, an electromotive force will be set up between the free ends of the wires. Its amount will depend upon the composition of the wires and the difference in temperature between the two. If a delicate galvanometer of high resistance be connected to the 'thermal couple,' as it is called, the deflection of the needle, after a careful calibration, will indicate the temperature very accurately.

"In the thermo-electric pyrometer of Le Chatelier, the wires used are platinum and a 10 per cent alloy of platinum and rhodium, enclosed in porcelain tubes to protect them from the oxidizing influence of the furnace gases. The couple with its protecting tubes is called an 'element.' The elements are made in different lengths to suit conditions.

"It is not necessary for accuracy to expose the whole length of the element to the temperature to be measured, as the electromotive force depends only upon the temperature of the juncture at the closed end of the protecting tube and that of the cold end of the element. The galvanometer can be located at any convenient point, since the length of the wires leading to it simply alter the resistance of the circuit, for which allowance may be made.

"The advantages of the thermo-electric pyrometer are accuracy over a wide range of temperatures, continuity of readings, and the ease with which observations can be taken. Its disadvantages are high first cost, and, in some cases, extreme delicacy."

For temperatures up to 3227° F., the fusing point of platinum, it is possible to make use of the melting points of various metals for approximate temperature indications.

Above these temperatures either *radiation* or *optical pyrometers* are employed, the former having a range as high as 3600° F., the limit in steam-boiler practice, and the latter being capable of recording temperatures as high as 12,000° F.

Measurement of Heat Quantity. (Calorimetry.) *Heat may be measured,* since it is a form of energy, in any of the usual energy units, as the joule, foot-pound, or horsepower hour. However, it is the custom to use for this purpose a special unit more readily applicable to heat changes. This unit in the English system is known as the **British thermal unit (B.t.u.),** and is the amount of heat required to raise 1 pound of water from 63° to 64° F.; in the French system the unit is called the *Calorie,* and is the amount of heat required to raise 1 kilogram of water from 15° to 16° C. Since 1 kg. = 2.2046 lb. and 1° C. = $^9/_5$° F., then 1 Cal. = 2.2046 × $^9/_5$ = 3.968 B.t.u. or 1 B.t.u. = 0.252 Cal.

The tendency at the present time is to define a B.t.u. as the mean or average amount of

heat per degree to raise 1 lb. of water from 32° to 212° F., which is almost exactly the same as the heat required to raise 1 lb. of water 1° at 63.5° F.

The *calorimeter* is an apparatus into which a hot body of known temperature and weight can be introduced, and in cooling through a known difference in temperature is made to give up heat measured in B.t.u. to a liquid also of known temperature and weight, which undergoes a corresponding increase in temperature.

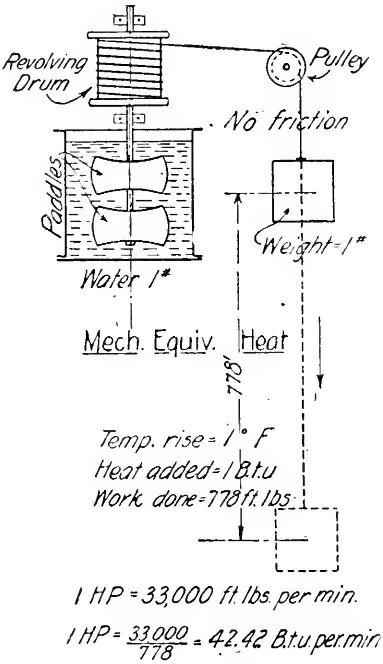


FIG. 8.

Example. If 1 lb. of iron is put into a calorimeter containing 10 lb. of water, and the water rises in temperature 5° F., the iron has given up 50 B.t.u., and at the same time its temperature has fallen about 420° F.

If 50 lb. of water are raised from 70° to 90° F. it is customary to say that 50 × (90–70) = 1,000 B.t.u. have been added.

It should be noted that while a B.t.u. is based on the temperature interval of 63° to 64° F. it will be sufficiently accurate for engineering work to use the actual temperature interval direct in any case without correction into terms of 63° to 64° F.

Specific Heat. It is a well-known fact that equal quantities of heat will raise equal weights of different substances a different number of degrees, depending on the nature of the substance. This property of matter is known as *specific heat*, and for any substance can be expressed as the number of B.t.u. required to raise or lower the temperature of 1 pound 1° F. at some given temperature. It is also customary to make use of the mean or average value for a certain temperature interval.

Two specific heats are recognized, one known as the "true" specific heat, measured at the temperature stated, and the other as the "mean" specific heat, which is the average value between the temperatures under consideration. In the case of gases a further distinction is made between specific heat at *constant pressure* and at *constant volume*. See "Specific Heat of Gases" in chapter on "Air."

The specific heat at constant pressure of a mixture of gases is obtained by multiplying the specific heat of each constituent gas by the percentage by weight of that gas in the mixture, and dividing the sum of the products by 100. The specific heat of a gas whose composition by weight is CO_2, 13 per cent; CO, 0.4 per cent; O, 8 per cent; N, 78.6 per cent, is found as follows:

$$
\begin{array}{rcrcl}
CO_2 & : & 13. \times 0.217 & = & 2.821 \\
CO & : & 0.4 \times 0.2479 & = & 0.09916 \\
O & : & 8. \times 0.2175 & = & 1.74000 \\
N & : & \underline{78.6} \times 0.2438 & = & \underline{19.16268} \\
 & & 100.0 \qquad & & 23.82284
\end{array}
$$

and 23.8228/100 = 0.238 = specific heat of the gas, at constant pressure.

The specific heats of various solids, liquids, and gases are given in Table 1.

Relation between Units of Energy and Power. Since the various forms of energy, heat, mechanical energy, electrical energy, etc., are mutually convertible there must be definite numerical relations between the various units used to express energy. As determined by various physicists the relation between the B.t.u. and ft.-lb. is

$$1 \text{ B.t.u.} = 777.64 \text{ ft.-lb.}$$

The number 777.64 is called the *mechanical equivalent of heat* and is denoted by J. For ordinary use the value 778 may be taken. Another convenient relation is, 1 hp.-hr. = 2,546 B.t.u.

TABLE 1

SPECIFIC HEATS OF VARIOUS SUBSTANCES

SOLIDS

	Temperature,* Degrees Fahrenheit	Specific Heat		Temperature,* Degrees Fahrenheit	Specific Heat
Copper	59–460	0.0951	Glass (normal ther. 16 III)	66–212	0.1988
Gold	32–212	.0316	Lead	59	.0299
Wrought iron	59–212	.1152	Platinum	32–212	.0323
Cast iron	68–212	.1189	Silver	32–212	.0559
Steel (soft)	68–208	.1175	Tin	−105–64	.0518
Steel (hard)	68–208	.1165	Ice	....	.5040
Zinc	32–212	.0935	Sulphur (newly fused)	....	.2025
Brass (yellow)	32	.0883			

LIQUIDS

	Temperature,* Degrees Fahrenheit	Specific Heat		Temperature,* Degrees Fahrenheit	Specific Heat
Water	59	1.0000	Sulphur (melted)	246–297	0.2350
Alcohol	32	0.5475	Tin (melted)	....	.637
	176	.7694	Sea-water (sp.gr.1.0043)	64	.980
Mercury	32	.3346	Sea-water (sp.gr.1.0463)	64	.903
Benzol	50	.4066	Oil of turpentine	32	.411
	122	.4502	Petroleum	64–210	.498
Glycerine	59–102	....	Sulphuric acid	68–133	.3363
Lead (melted)	to 360	.0410	Olive oil	....	.309

GASES

	Temperature,* Degrees Fahrenheit	Specific Heat at Constant Pressure	Specific Heat at Constant Volume		Temperature,* Degrees Fahrenheit	Specific Heat at Constant Pressure	Specific Heat at Constant Volume
Air	32–392	0.2375	0.1693	Carbon monoxide	41–208	0.2425	0.1728
Oxygen	55–405	.2175	.1553	Carbon dioxide	52–417	.2169	.1535
Nitrogen	32–392	.2438	.1729	Methane	64–406	.5929	.4505
Hydrogen	54–388	3.4090	2.4141	Blast-Fur. gas (approx.)	....	.2277	.....
				Flue gas (approx.)	....	.2400	.....

SPECIFIC HEAT OF BUILDING MATERIALS

Building Materials	Specific Heat	Building Materials	Specific Heat	Densities	Lb. per 1 Cu. Ft.
Brick work	0.1950	Oakwood	0.5700	Stone work	160
Masonry	.2159	Birch	.4800	Wood	40
Plaster	.2000	Glass	.1977	Slate	170
Pinewood	.4670			Plaster	90

* When one temperature alone is given the "true" specific heat is given; otherwise the value is the "mean" specific heat for the range of temperature given.

One method used for determining the value of J is shown diagramatically in Fig. 8. This apparatus consisted essentially of a paddle-wheel revolved by a cord wound around a drum and connected to a known weight which in falling through a known distance caused the wheel to stir up the water and thus transmit the energy of the falling weight to the paddle. The friction of the water against this wheel produces heat which raises the temperature of the water a known number of degrees.

The unit of electrical energy is the *joule*, and the corresponding unit of power is the *watt*,

or one watt is the same as one joule per second. The larger unit of power is the kilowatt (kw.) = 1,000 watts. The following are the relations between these units and other units, or the *electrical equivalents of heat.*

1 watt-hour = 3.415 B.t.u.
1 kw.-hour = 3,415 B.t.u.
1 hp.-hour = 746 watts = 0.746 kw.

The numerical relations between the various units of pressure, energy, and power is given in the following table.

TABLE 2

EQUIVALENT VALUES OF ELECTRICAL AND MECHANICAL UNITS
(*H. Ward Leonard* in "The Electrical Engineer," February 25, 1895, Revised.)

Unit	Equivalent Value in Other Units
1 Kilowatt hour.....	1,000 watt hours 1.34 h.p. hours 2,654,200 ft. lb. 3,600,000 joules 3,414.5 B.t.u. 367,100 kilogram meters 0.235 lb. carbon oxidized with perfect efficiency 3.53 lb. water evaporated from and at 212° F. 22.75 lb. of water raised from 62° to 212° F.
1 H.P. hour........	0.746 kw. hours 1,980,000 ft. lb. 2,546 B.t.u. 273,750 kg. m. 0.175 lb. carbon oxidized with perfect efficiency 2.64 lb. water evaporated from and at 212° F. 17.0 lb. water raised from 62° F. to 212° F.
1 Kilowatt..........	1,000 watts 1.34 horse-power 2,654,200 ft. lb. per hour 44,240 ft. lb. per minute 737.6 ft. lb. per second 3,414.5 B.t.u. per hour 56.9 B.t.u. per minute 0.948 B.t.u. per second 0.2275 lb. carbon oxidized per hour 3.53 lb. water evaporated per hour from and at 212° F.
1 Horsepower.......	746 watts 0.746 kw. 33,000 ft. lb. per minute 550 ft. lb. per second 2,546 B.t.u. per hour 42.4 B.t.u. per minute 0.707 B.t.u. per second 0.175 lb. carbon oxidized per hour 2.64 lb. of water evaporated per hour from and at 212° F.

Sensible and Latent Heat. Whenever we add heat to a substance without change of state we increase its temperature, and the heat thus added is known as *sensible heat*, as, for example, the heat added to water between 50° and 140° F. Sensible heat changes, as already stated, are measured by the thermometer.

Heat may be added to a body without any change of temperature provided a change of state from solid to liquid or from liquid to vapor takes place, and the heat thus added is known as *latent heat*. When the change is from solid to liquid, as ice to water, this heat is known as the *latent heat of fusion*. At atmospheric pressure ice melts at 32° F. and the latent heat is 144 B.t.u. per pound.

When the change is from liquid to vapor, as water to steam, the heat required to effect the change is known as the *latent heat of evaporation*. At atmospheric pressure water evaporates at 212° F. and the latent heat is 971.7 B.t.u. per pound.

TABLE 3

APPROXIMATE MELTING POINTS OF METALS AND OTHER SUBSTANCES

Metal or Other Substance	Temperature, Deg. Fahrenheit	Metal or Other Substance	Temperature, Deg. Fahrenheit
Wrought iron	2737	Lead	621
Pig iron (gray)	2190–2327	Bismuth	498
Cast iron (white)	2075	Tin	449
Steel	2460–2550	Platinum	3191
Steel (cast)	2500	Gold	1946
Copper	1981	Silver	1762
Zinc	786	Aluminum	1216
Antimony	1166	Mercury	−39
Ice	32	Carbon dioxide	−108
Tallow	92	Sulphur dioxide	−148
Stearic acid	158		
Sulphur	239		

In no case is this latent heat lost, as it always reappears whenever the substance passes through the reverse process from gas or vapor to liquid or from liquid to solid.

The temperature of ebullition of any liquid, or the *boiling-point*, may be defined as the temperature which exists when the addition of heat to the liquid no longer causes rise of temperature, the heat added being absorbed or utilized in converting the liquid into vapor. This temperature is dependent upon the pressure under which the liquid is evaporated, being higher as the pressure is greater. See Table 5 in chapter on "Water, Steam, and Air."

Expansion of Solids. The addition of heat, to practically all substances, causes them to expand or increase in length, area, and volume, providing no change of state takes place during heating. The amount by which one unit of length, area, or volume of the substance changes in length, area, or volume per 1° rise in temperature is known as the *coefficient of linear, superficial, or cubical expansion,* respectively. The coefficient of expansion is not a constant quantity and hence the temperature range to which the coefficient applies should always be stated. The variation is slight for the same material and the coefficient is usually assumed constant for any given substance.

TABLE 4

LINEAL EXPANSION OF SOLIDS AT ORDINARY TEMPERATURES

(Tabular values represent increase per foot per 100 degrees increase in temperature, Fahrenheit)

Substance	Temperature Conditions,* Degrees Fahrenheit	Coefficient per 100 Degrees Fahrenheit
Brass (cast)	32 to 212	0.001042
Brass (wire)	32 to 212	.001072
Copper	32 to 212	.000926
Glass (English flint)	32 to 212	.000451
Granite (average)	32 to 212	.000482
Iron (cast)	104	.000589
Iron (soft forged)	0 to 212	.000634
Iron (wire)	32 to 212	.000800
Lead	32 to 212	.001505
Mercury†	32 to 212	.009984
Limestone	32 to 212	.000139
Steel (Bessemer rolled, hard)	0 to 212	.00056
Steel (Bessemer rolled, soft)	0 to 212	.00063
Steel (cast, French)	104	.000734
Steel (cast annealed, English)	104	.000608

* Where range of temperature is given, coefficient is mean over range.
† Coefficient of cubical expansion.

Propagation of Heat. Heat may be propagated by conduction, convection, and radiation.

Conduction is a molecular transmission of heat, the material in question transmitting the heat from particle to particle of its own substance. This transmission will only occur between

any two sections of the material which are at different temperatures, the heat always flowing from the higher to the lower temperature.

Time is required for conduction to take place, and varies with the distance between the sections, with the temperature difference, and with the character of the material. Good conductors, permit a very rapid flow, while poor conductors transmit heat very slowly. In these latter substances great differences of temperature may exist, while in the former the substance arrives at very nearly the same temperature throughout in a very short time.

Since conduction takes place between molecules by contact it may go on *in any direction* from the source of heat, and hence does not always travel in straight-lines like radiation. The amount of heat which is transmitted per unit of time by conduction is directly proportional to the area of the cross-section, to the difference of the temperatures divided by the thickness, and to a coefficient which depends on the character of the material.

The *coefficient of conduction* is the quantity of heat which flows in unit time, through a cross-section of unit area, when the thickness of the plate is unity and the difference of temperature is one degree. In the English system the relation that determines this coefficient is

$$Q = C\,S\,\frac{(t_2 - t_1)}{X}\,T$$

Q = quantity of heat in B.t.u. C = coefficient of conduction per 1 in. thickness, S = area in sq. ft. X = thickness in inches, $t_2 - t_1$ = the temperature difference between the two sections or surfaces, and T = time in hours.

The conducting power of substances varies greatly, as shown by the table of absolute conductivities of various materials in the chapter on "Heat Transmission of Building Materials."

Convection is the transmission of heat by the circulation of one substance, a fluid or gas, over the surface of a hotter or colder body. The particles or molecules of the moving substance come into close contact with the hotter body, and are actually heated by conduction during the period of this contact, but immediately pass on, carrying what heat they have acquired along with them, and fresh, cooler molecules succeed them. This circulation may be caused by purely natural forces, or may be produced by mechanical means. The circulation of the water in a boiler is an example of the former, while the circulation of air over the heater coils in a fan blast heating system is an example of the latter condition. In case the circulating substance is hotter than the other body the process will be reversed and heat will be given up by the moving molecules.

In general, it may be said that the heat transferred by convection is independent of the nature of the surface of the body and of the surrounding absolute temperature. It depends on the velocity of the moving substance, varying as some function of the velocity, on the form and dimensions of the body; and on the temperature difference between the moving substance and the body.

The general expression for the *heat given off by convection* is:

$$H = f\,(V)^{\frac{1}{n}}\,(t_s - t_a)\,A,$$

where V is the velocity in feet per second, t_s and t_a are the temperatures of the heating media and outside air respectively in degrees F., A is the area in sq. ft., and f and n are constants to be determined for the radiator in question.

Radiation is the transmission of heat through a medium commonly known as the ether, which is assumed to occupy all intermolecular spaces. Radiation always takes place *in straight lines*, obeying the same laws as light, so that its intensity or amount per unit of surface varies inversely as the square of the distance from the source of radiation to the surface, and directly with the sine of the angle of inclination. Moreover, radiant heat continues to travel in the same straight line until intercepted or absorbed by some other body.

TABLE 5

RELATIVE RADIATING OR ABSORBING POWER AT 212° F.

Lampblack	100	Steel	17
White lead	100	Platinum	17
Paper	98	Polished brass	7
Glass	90	Copper	7
India ink	85	Polished gold	3
Shellac	72	" silver	3

The amount of radiant heat emitted or absorbed depends largely upon the character of the surface of the hot or cold body, and it has been found that the power of a given substance for absorbing radiant heat is exactly the same as for emitting radiant heat. Table 5 gives the relative radiating powers of various substances at 212° F.

Radiant heat has the property of *passing through dry gases without heating them* to any appreciable extent, but air containing water vapor or dust will intercept and absorb radiant heat, hence the earth's atmosphere is warmed by the radiant heat from the sun.

Radiant heat like light is reflected from various materials, and it will be found that in general substances possessing a high power of radiation have a low reflecting power. Silver has a relative radiating power of 3 but its reflecting power is given as 97.

Radiant heat will also pass through certain solid substances without heating them, in the same way as light passes through glass. This property of substances is known as *diathermacy*, and crystals of rock salt have this property to a very high degree.

Radiant heat is diffused in all directions by certain materials such as white lead, powdered silver, and chromate of lead. Radiation in this case takes place in all directions, with little or no regularity or uniformity of direction.

In general it may be said that the heat emitted by radiation per unit of surface and per unit of time is independent of the form and extent of the heated body provided there are no re-entrant surfaces to intercept the heat rays. Also, the amount of heat emitted by a surface radiating equally in all directions depends only on the nature of the surface, the difference in temperature between the surface and surroundings, and the absolute value of the temperature.

The general expression for *heat given off by radiation*, as stated by *Newton*, and later by *Dulong* and *Petit*, as well as *Stefan* and *Boltzman*, is:

$$H = K\left(T_1^x - T_2^x\right),$$

where K is the radiation constant or coefficient, and T_1 and T_2 are the absolute temperatures of the hot body and the surrounding colder bodies respectively. *Newton* gave the exponent x the value 1, but this has since been proved too small, and *Stefan* and *Boltzman* give the value $x = 4$, while for a black body they give $K = (16 \times 10^{-10})$.

CHAPTER II

WATER, STEAM, AND AIR

WATER

Properties of Water. Pure water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen gas with one volume of oxygen gas, or 2 parts by weight of hydrogen and 16 parts by weight of oxygen. Water expands when heated from 39.2° F., the temperature of maximum density, to any higher temperature, but contracts when heated from 32° to 39.2° F. At the atmospheric pressure of 29.92″ mercury its freezing point is 32° F. and its boiling point is 212° F.

The change in density is shown by the following comparison of weights per cu. ft. at various temperatures.

At	32° F., or freezing point	=	62.418 lb. per cu. ft.
"	39.2° F., max. density	=	62.427 " " " "
"	62° F., standard	=	62.355 " " " "
"	212° F., or boiling point	=	59.760 " " " "

TABLE 1

HEAT CONTENT AND SPECIFIC WEIGHT OF WATER

Temp., Deg. Fahr.	Heat Content Above 32° per 1 Lb.	Weight, Lb. per Cu. Ft.	Temp., Deg. Fahr.	Heat Content Above 32° per 1 Lb.	Weight, Lb. per Cu. Ft.	Temp., Deg. Fahr.	Heat Content Above 32° per 1 Lb.	Weight, Lb. per Cu. Ft.
32	0.00	62.42	100	68.00	62.02	158	125.88	61.02
35	3.02	62.42	102	69.99	62.00	160	127.88	60.98
40	8.05	62.42	104	71.99	61.97	162	129.87	60.94
45	13.07	62.42	106	73.98	61.95	164	131.87	60.90
50	18.08	62.41	108	75.97	61.92	166	133.86	60.85
52	20.08	62.40	110	77.97	61.89	168	135.88	60.81
54	22.08	62.40	112	79.96	61.86	170	137.88	60.77
56	24.08	62.39	114	81.96	61.83	172	139.88	60.73
58	26.08	62.38	116	83.95	61.80	174	141.88	60.68
60	28.08	62.37	118	85.94	61.77	176	143.89	60.64
62	30.08	62.36	120	87.94	61.74	178	145.89	60.59
64	32.08	62.35	122	89.93	61.70	180	147.89	60.55
66	34.08	62.34	124	91.93	61.67	182	149.89	60.50
68	36.08	62.33	126	93.92	61.63	184	151.90	60.46
70	38.07	62.31	128	95.92	61.60	186	153.90	60.41
72	40.07	62.30	130	97.91	61.56	188	155.91	60.37
74	42.07	62.28	132	99.91	61.52	190	157.91	60.32
76	44.06	62.27	134	101.90	61.49	192	159.92	60.27
78	46.06	62.25	136	103.90	61.45	194	161.92	60.22
80	48.05	62.23	138	105.90	61.41	196	163.93	60.17
82	50.05	62.21	140	107.89	61.37	198	165.94	60.12
84	52.04	62.19	142	109.89	61.34	200	167.95	60.07
86	54.04	62.17	144	111.89	61.30	202	169.95	60.02
88	56.03	62.15	146	113.89	61.26	204	171.96	59.97
90	58.03	62.13	148	115.88	61.22	206	173.97	59.92
92	60.02	62.11	150	117.88	61.18	208	175.98	59.87
94	62.02	62.09	152	119.88	61.14	210	177.99	59.82
96	64.01	62.07	154	121.88	61.10	212	180.00	59.76
98	66.01	62.05	156	123.88	61.06			

At 62° a U. S. gallon of 231 cu. in. weighs approximately 8⅓ lb., and a cu. ft. is equal to 7.48 gals. Pressures are often stated in feet or inches of water column, and at 62° F. the equiva-

lent in pounds per sq. ft. is (let h = head in feet), = 62.355 h, or in pounds per sq. in. = $\frac{62.355}{144} h = 0.433\ h$. Also, if h_1 = head in inches of water at 62° F., then the pressure in ounces per sq. in. $= \frac{h_1}{12} \frac{62.355}{144} \times 16 = 0.578\ h_1$, or $h_1 = 1.73 \times$ pressure in ounces per sq. in. A column of water 2.309 ft. or 27.71 in. high exerts a pressure of 1 lb. per sq. in. at 62° F.

For density of water at other temperatures than those already stated see Table 1.

The *specific volume* of water, or the volume of one pound, depends on the temperature at which the volume is measured, and is practically independent of the pressure, since water is but very slightly compressible. The specific volume is the reciprocal of the specific density, values for the latter being given in Table 1, hence it is only necessary to find the value of $\frac{1}{\text{wt. per cu. ft.}}$ to get the volume of 1 pound, as $\frac{1}{62.42} = 0.016$ cu. ft. at 32° F.

The *boiling point* of pure water varies with the pressure or altitude above sea level, the temperature at which ebullition will occur decreasing with the altitude or lower pressure. This relation is shown by reference to the steam tables, which also indicate that the boiling point increases for pressures higher than that of the atmosphere at sea level. Table 2 gives the boiling points at various altitudes.

TABLE 2

BOILING POINT OF WATER AT VARIOUS ALTITUDES

Boiling Point, Degrees Fahr.	Altitude Above Sea Level, Feet	Atmospheric Pressure, Pounds per Sq. In.	Barometer Reduced to 32 Degrees, Inches	Boiling Point, Degree Fahr.	Altitude Above Sea Level, Feet	Atmospheric Pressure, Pounds per Sq. In.	Barometer Reduced to 32 Degrees, Inches
184	15,221	8.20	16.70	199	6,843	11.29	22.99
185	14,649	8.38	17.06	200	6,304	11.52	23.47
186	14,075	8.57	17.45	201	5,764	11.76	23.95
187	13,498	8.76	17.83	202	5,225	12.01	24.45
188	12,934	8.95	18.22	203	4,697	12.26	24.96
189	12,367	9.14	18.61	204	4,169	12.51	25.48
190	11,799	9.34	19.02	205	3,642	12.77	26.00
191	11,243	9.54	19.43	206	3,115	13.03	26.53
192	10,685	9.74	19.85	207	2,589	13.30	27.08
193	10,127	9.95	20.27	208	2,063	13.57	27.63
194	9,579	10.17	20.71	209	1,539	13.85	28.19
195	9,031	10.39	21.15	210	1,025	14.13	28.76
196	8,481	10.61	21.60	211	512	14.41	29.33
197	7,932	10.83	22.05	212	Sea Level	14.70	29.92
198	7,381	11.06	22.52				

The *specific heat of water*, or the number of B.t.u. required to raise the temperature of 1 pound of water 1° F. varies with the temperature as shown in the following table.

TABLE 3

Temperature, F.°	Specific Heat
30° F.	1.0098
55	1.0000
100	0.9967
160	1.0002
210	1.0050

In consequence of this variation, the amount of heat required to raise 1 lb. of water at 32° F. through a known temperature interval, known as the *heat of the liquid*, will depend on the average value of the specific heat for that range, and this variation is shown in Table 1—where the "heat units" required to raise 1 lb. of water from 32° F. to the temperature in the table is given as the heat content.

The specific heat of water is very commonly assumed to be unity, and is so used in many engineering calculations. The steam tables, however, are based on the exact value for the temperature range in question.

The specific heat of ice at 32° F. is 0.463 B.t.u. per 1 pound.

Flow of Water in Pipes. The flow of water in pipes depends on a difference in head or pressure between the two points between which flow takes place. This difference in head is used up in overcoming the resistance (friction of the pipe) offered to the flow, and in creating the velocity of discharge at the second point.

The flow of a liquid in a pipe is under the influence of three heads or equivalent pressures.

The *velocity head or pressure* is defined as that head or pressure of the liquid which is required to create the velocity of flow, that is, the head or pressure necessary to accelerate the mass from a state of rest to the velocity attained at the point in the line under consideration.

The *resistance head* or *pressure*, also termed the *friction head*, is that head or pressure required to overcome the frictional resistance offered to the flow.

The *total head or pressure*, also termed the *dynamic head or pressure*, is the sum of the velocity head plus the friction head.

The *potential or measured head* is the vertical distance, measured in feet, from some datum line to the center of the pipe at the point in the line under consideration.

The Piezometer in its simplest form consists of a tube inserted in a pipe at right angles to the flow (Fig. 1). The radial pressure within the pipe is measured by the height of the column of the liquid within the tube. For high pressures an ordinary gage of the Bourdon type is substituted for the tube.

The reading obtained by the use of a piezometer placed in a pipe of uniform cross section throughout its entire length with free discharge to the atmosphere is the head lost by friction beyond the point of attachment of the piezometer.

The Pitot Tube in its simplest form is a bent tube placed in the pipe so that the immersed end of the tube faces the stream (Fig. 2). The height of the column of liquid in the tube is greater

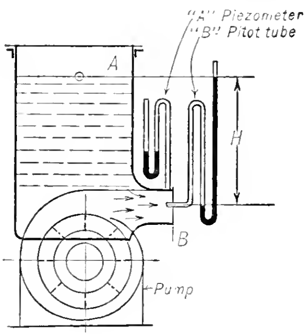


FIG. 1.

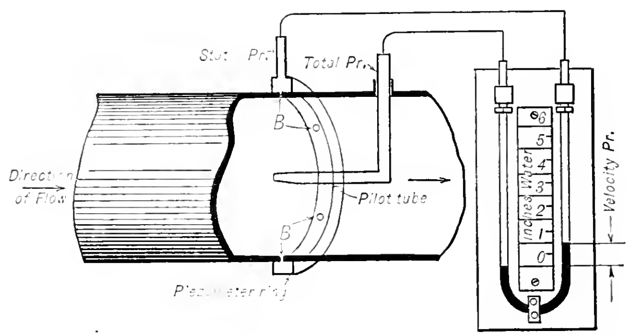


FIG. 2.

than the reading obtained by the piezometer by an amount equal to the head required to produce the velocity of flow. The height of the column is the *total head* at the point of measurement.

The *difference* between the readings obtained by the Pitot tube and the piezometer is the velocity head at the point considered. If the pipe is of uniform diameter this difference is of course a constant throughout the length of the pipe as the velocity is constant.

The difference between the total and resistance heads is read direct on the manometer by connecting the opposite ends of the U tube to the piezometer and Pitot tube as shown by Fig. 2.

If it were not for friction "the total head at any point or section would be equal to the total

head at any subsequent point or section," total head being the sum of the static or friction head plus the velocity head. See Fig. 4. $H = h_s + h_v$, where

H = total head measured in feet of fluid flowing.

h_s = static or friction head measured in feet of fluid flowing.

h_v = velocity head measured in feet of fluid flowing.

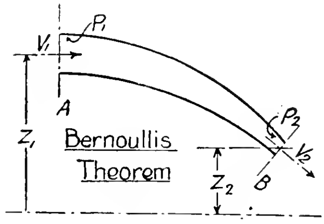


FIG. 3.

This relation between the total head at any two sections of a pipe line, assuming frictionless flow, is known as *Bernoulli's theorem* and is demonstrated as follows. See Fig. 3.

Assume (1) a perfect fluid, (2) steady flow, (3) no friction. Assume a weight W passes section A in unit time. Because of (2) a weight W also passes B in the same time.

$$\text{Kinetic energy of } W \text{ at } A = \tfrac{1}{2} M V^{2*} \left(\text{where } M = \frac{W}{g}\right)$$

Let P_1 = the radial or static pressure at section A measured in pounds per sq. ft. and P_2 the static pressure at B measured in pounds per sq. ft.

Potential energy of W at $A = WZ_1 + \dfrac{P_1}{D} W$ in which D is the density of the fluid flowing hence $\dfrac{P_1}{D}$ is the potential head equivalent to the static pressure P_1, and Z_1 = potential, head or measured

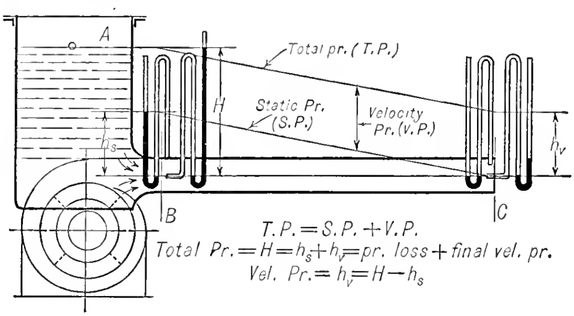


FIG. 4.

head at the section. The total energy of W at $A = W\left(\dfrac{V_1^2}{2g} + Z_1 + \dfrac{P_1}{D}\right)$. Likewise, the total energy of W at $B = W\left(\dfrac{V_2^2}{2g} + Z_2 + \dfrac{P_2}{D}\right)$. Since there is no external frictional resistance the total energy at A equals that at B or

$$\frac{V_1^2}{2g} + Z_1 + \frac{P_1}{D} = \frac{V_2^2}{2g} + Z_2 + \frac{P_2}{D}$$

This is Bernoulli's theorem, and each member of the equation is the "total head" at the corresponding section. It may be stated thus: In a steady flow *without friction* the total head

* M has a velocity V, and a constant force F would bring it to rest in a time t, and a distance S with a negative acceleration a, $S = \tfrac{1}{2} at^2$ and $F = Ma$. The work obtained (i.e. the kinetic energy of M) equals $F \times S = \tfrac{1}{2} M \times a^2t^2$. But $v = at$. $\therefore$ Kinetic energy of $M = \tfrac{1}{2} M V_1^2 = \dfrac{W V_1^2}{2g}$.

at any section equals the total head at any subsequent section. Note that the "total head" is the sum of the "velocity" head, the "potential" head, and the "pressure" head.

Case I. (*Flow without friction*): Apply Bernoulli's theorem to the case of water issuing from the base of a stand pipe. See Fig. 1.

The pressure at A is atmospheric (P_a) and within the jet at B it is also atmospheric. The velocity at A is zero

$$0 + H + \frac{P_a}{D} = \frac{V^2}{2g} + 0 + \frac{P_a}{D}$$

$$\therefore H = \frac{V^2}{2g} \text{ or } V = \sqrt{2gH}$$

Case II. (*Flow with friction*): Since friction tends to oppose motion the total head at any section is greater than the total head at any subsequent section. The "lost" or "friction" head

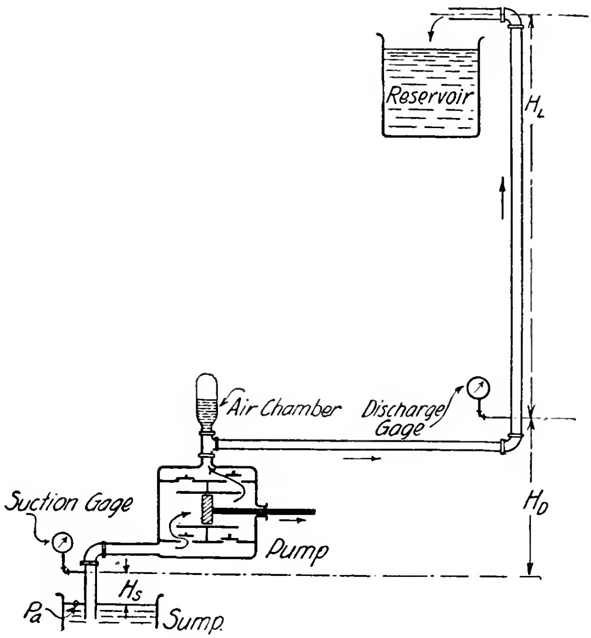


FIG. 5.

between any two sections is therefore the difference between the total heads at these sections. (Fig. 4.)

The total head at $A = 0 + H + \frac{P_a}{D}$

The total head at $C = \frac{V^2}{2g} + 0 + \frac{P_a}{D}$

The friction or lost head $= \left(0 + H + \frac{P_a}{D}\right) - \left(\frac{V^2}{2g} + 0 + \frac{P_a}{D}\right) = H - \frac{V^2}{2g}$

By applying the equation between A and B, or B and C it can readily be seen that the "pressure" or "static" head at B equals the friction or lost head caused by the pipe line.

Application of Bernoulli's theorem to the case of a pump to show what the suction and discharge gages on the pump register, and to show how the "total" head on a pump may be found. (Fig. 5.) Call H_T the "total" head on pump, h_1 the friction head lost in the suction pipe, h_2 the friction head lost in the discharge pipe, V_s the suction velocity, and V_d the discharge velocity.

Applying the equation between the surface of the sump and the suction gage:

$$\left(0 + 0 + \frac{P_a}{D}\right) - h_1 = \left(\frac{V^2_s}{2g} + H_s + \frac{P_s}{D}\right) \quad \ldots\ldots\ldots \quad (1)$$

The pressure registered on the suction gage is $(P_a - P_s)$ where P_s is the absolute pressure at this section. From equation (1) $\frac{1}{D}(P_a - P_s) = \left(H_s + \frac{V^2_s}{2g} + h_1\right)$.

The head registered on the suction gage equals the suction lift, plus the suction velocity head, plus the suction friction head.

Applying the equation between the discharge gage and the end of the line:

$$\left(\frac{V^2d}{2g} + 0 + \frac{Pd}{D}\right) - h_2 = \left(\frac{V^2d}{2g} + H_L + \frac{P_a}{D}\right)$$

$$\therefore \frac{1}{D}(P_D - P_a) = h_2 + H_L$$

The discharge gage registers the friction head in the discharge line, plus the measured lift of the discharge line.

Apply the theorem between the surface of the sump and the discharge pressure gage. (Note that the head H_T is added to the water during its passage through the pump.)

$$\left(0 + 0 + \frac{P_a}{D}\right) - h_1 + H_T = \frac{V^2d}{2g} + H_D + H_s + \frac{Pd}{D}$$

$$\therefore \left(\frac{Pd}{D} - \frac{P_a}{D}\right) = H_T - h_1 - H_D - H_s - \frac{V^2d}{2g}$$

or

$$H_T = \left(\frac{Pd}{D} - \frac{P_a}{D}\right) + H_s + h_1 + \frac{V^2d}{2g} + H_D \quad \ldots\ldots \quad (2)$$

but

$$\left(\frac{Pd}{D} - \frac{P_a}{D}\right) = h_2 + H_L$$

$$\therefore H_T = h_2 + h_1 + H_s + H_D + H_L + \frac{V^2d}{2g}$$

The *total head on the pump is equal to the entire friction head plus the measured head plus the final discharge velocity head.*

When the head is produced on the pump by closing the discharge valve, the measured head does not exist in reality but only virtually. The total head must be found from the two gage readings, the velocities in the suction and discharge lines, and the distance between gages.

From equation (2)

$$H_T = \left(\frac{Pd}{D} - \frac{P_a}{D}\right) + \left(H_s + h_1 + \frac{V^2_s}{2g}\right) + \frac{V^2d}{2g} - \frac{V^2_s}{2g} + H_D$$

$$= \left(\frac{Pd}{D} - \frac{P_a}{D}\right) + \left(\frac{P_a}{D} - \frac{P_s}{D}\right) + \frac{V^2d}{2g} - \frac{V^2_s}{2g} + H_D$$

The total head equals the discharge pressure head plus the suction pressure head plus the final velocity head minus the suction velocity head plus the distance between gages.

If the size of the discharge pipe equals that of the suction pipe the total head is found more easily.

V_d will equal V_s

Substituting $\frac{V^2_s}{2g}$ for $\frac{V^2_d}{2g}$ in (2)

$$H_T = \left(\frac{P_d}{D} - \frac{P_a}{D}\right) + \left(H_s + h_1 + \frac{V^2_s}{2g}\right) + H_D$$

$$H_T = \left(\frac{P_d}{D} - \frac{P_a}{D}\right) + \left(\frac{P_a}{D} - \frac{P_s}{D}\right) + H_D$$

The *total head on the pump equals the discharge pressure head plus the suction pressure head plus the distance between gages.*

Friction Head due to Flow of Water in Pipes. The *flow of water* in a pipe of uniform diameter will take place with a constant velocity if the total head producing flow is maintained constant This total head can be determined for any given velocity of flow if the friction head is known.

The loss of head due to friction when a fluid such as water, steam, air, or gas flows through a straight tube or pipe is generally represented by the formula,

$$h = f\frac{L\,R}{A}\,\frac{v^2}{2g}$$

where f = the coefficient of friction. L = length of tube in feet; R = perimeter of tube in feet, A = area in sq. ft., v = velocity of flow in feet per sec., and h = friction head in feet of the fluid flowing.

If the tube is round and D = diameter in feet, then $h = f\frac{\pi\,D\,L}{\frac{\pi\,D^2}{4}}\,\frac{v^2}{2g} = f\,\frac{4\,L}{D}\,\frac{v^2}{2g}$ in which

f = .00644 according to *Weisbach*, for clean iron pipe.

This formula may be reduced to $h = f\frac{2\,L}{D}\,\frac{v^2}{g}$ or $h = f_1\frac{L}{D}\,\frac{v^2}{2g}$ in which f_1 = 0.02, an average for water.

It is understood that the pipe is smooth, clean and free from the burrs as ordinarily left by a wheel pipe cutter.

For very low velocities, as found in gravity hot water heating systems, the above formula does not hold good.

William Cox in the "American Machinist," Dec. 28, 1913, gives the following modification of the above formula, which is simpler and gives almost identical results.

$$h = \frac{L}{d}\,\frac{(4\,v^2 + 5\,v - 2)}{1200}.$$

Values of the expression $\frac{(4\,v^2 + 5\,v - 2)}{1200}$ can be tabulated for varying velocities so that h may be readily solved for when v, L, and d are known. See Table 4, for these tabulated values. In Cox's formula d = diameter in inches.

TABLE 4

VALUES OF $\frac{4v^2 + 5v - 2}{1200}$

v	0.0	0.2	0.4	0.6	0.8
1	0.00583	0.00813	0.01070	0.01353	0.01663
2	.02000	.02363	.02753	.03170	.03613
3	.04083	.04580	.05103	.05653	.06230
4	.06833	.07463	.08120	.08803	.09513
5	0.10250	0.11013	0.11803	0.12620	0.13463
6	.14333	.15230	.16153	.17103	.18080
7	.19083	.20113	.21170	.22253	.22363
8	.24500	.25663	.26853	.28070	.29313
9	0.30583	0.31880	0.33203	0.34553	0.35930
10	.37333	.38763	.40220	.41703	.43213
11	.44750	.46313	.47903	.49520	.51163
12	.52833	.54530	.56253	.58003	.59780
13	0.61583	0.63413	0.65270	0.67153	0.69063
14	.71000	.72963	.74953	.76970	.79013
15	.81083	.83180	.85303	.87453	.89630
16	.91833	.94063	.96320	.98603	1.00913
17	1.03250	1.05613	1.08003	1.10420	1.12863
18	1.15333	1.17830	1.20353	1.22903	1.25480
19	1.28083	1.30713	1.33370	1.36053	1.38763
20	1.41500	1.44263	1.47053	1.49870	1.52713
21	1.55583	1.58480	1.61403	1.64353	1.67330

The use of the formula and table may be illustrated as follows:

Example. Given a pipe 5″ in diameter and 1,000 ft. long, with 49 ft. head, what will be the discharge?

If the velocity v is known in feet per second, the discharge will be $\pi \frac{d^2}{4} \times \frac{60}{144} \times v = 0.32725\, d^2v$ cu. ft. per min. $= Q$. Now $\frac{hd}{L} = \frac{49 \times 5}{1000} = \frac{4v^2+5v-2}{1200} = 0.245$ and by reference to the table it will be seen that the actual velocity $v = 8$ ft. per sec.

The discharge in cu. ft. per min., if v is velocity in feet per second and d the diameter in inches is $0.32725\ d^2v$, hence $Q = 0.32725 \times 25 \times 8 = 65.45$ cu. ft. per min.

The velocity due to the head, if there were no friction, is $8.025\ \sqrt{h} = 56.175$ ft. per sec. and the discharge at that velocity would be $0.32725 \times 25 \times 56.175 = 460$ cu. ft. per min.

Example. Suppose it is required to deliver this amount, 460 cu. ft., at a velocity of 2 ft. per sec.; what diameter of pipe of the same length and under the same head will be required and what will be the loss of head by friction?

$$d = \text{diameter} = \sqrt{\frac{Q}{v \times 0.32725}} = \sqrt{\frac{460}{2 \times 0.32725}} = \sqrt{703} = 26.5 \text{ inches diameter.}$$

Since the diameter, velocity and discharge are now known the friction head is found from

$$h = \frac{L}{d} \times \frac{(4v^2+5v-2)}{1200} \text{ using the table; thus,}$$

$$h = \frac{1000}{26.5} \times 0.02 = \frac{20}{26.5} = 0.75 \text{ ft.}$$

Friction Pressure Loss Chart for Flow of Water. The *chart* (Fig. 6) from the "American Machinist," is based on the preceding formula. It gives the *velocity of flow* in pipes of various nominal diameters, and also the *friction or pressure loss* in pounds per sq. in. per 100 ft. of pipe, at varying rates of flow, stated both in gallons per min., and in cu. ft. per min.

The corresponding velocity of flow in lineal feet per sec. is read from the same chart by referring to the velocity lines, which in the example given on the sheet would be 5.9 ft. per sec.

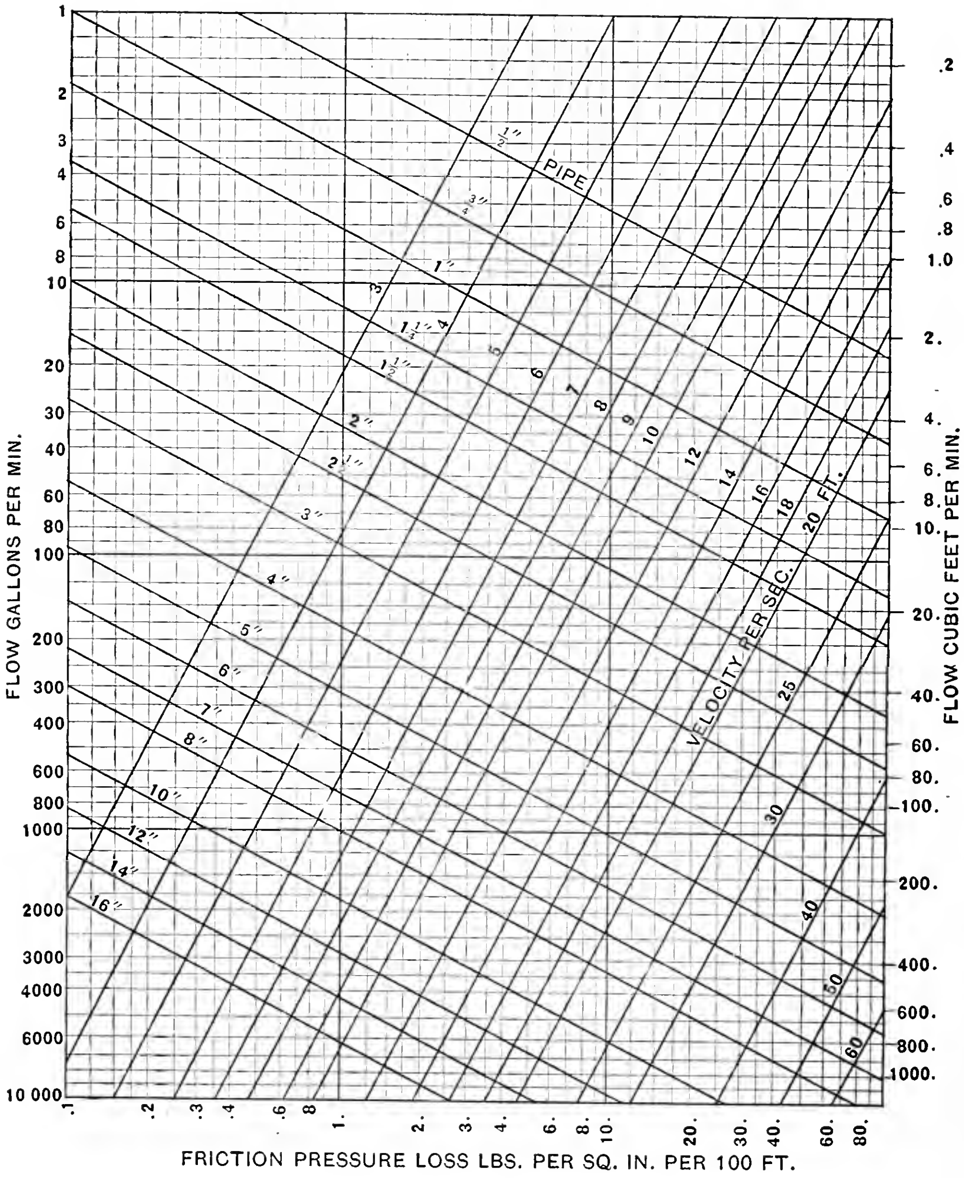


FIG. 6. FLOW OF WATER IN PIPES.

Example. 60 gals. per min. to be transmitted 300 ft. through a 2″ standard steel pipe. Required the friction loss. From 60 gals. on the left trace horizontally to the intersection with the diagonal 2″ pipe, and read 3.25 lb. per sq. in. at the bottom of the chart. The loss is then $3 \times 3.25 = 9.75$ lb. per sq. in.

Approximate Allowance for Ells and Globe Valves.

Add to the measured length of line 40 diams. for each 90° ell, and 60 diams. for each globe valve.

Loss of Head by Entrance, Elbows and Valves. The loss of head occasioned by entrance to a pipe and various obstructions may be stated as a function of the velocity head as $h = \phi \frac{V^2}{2g}$ in which ϕ is a coefficient experimentally determined.

Values of ϕ. This may be taken to equal 0.50 for a pipe at right angles to the reservoir where the pipe is flush with the inside surface with the burr removed so that the edge is sharp. Approximately the same condition exists when a smaller branch pipe is taken off a main.

When the pipe projects inside the reservoir for a length equal to several diameters the value of ϕ may be taken as 0.93. If the entrance is bell-mouthed and smooth the value of ϕ may be practically equal to 0.

The value of ϕ for elbows as stated by *Weisbach* based on experiments conducted with 1¼-inch pipe are as follows:

Angle of elbow	= 22½°	45°	90°
Value of ϕ	= 0.038	0.181	0.984

For smaller pipe the value of ϕ increases. For example, *Weisbach* gives ϕ = 1.53 for a 90 degree ⅜-inch elbow. For larger pipe the value of ϕ becomes less.

The value of ϕ for a globe valve, wide open, is ordinarily assumed as 1.5 times the value for a 90 degree elbow. The loss through a gate valve, wide open, is ordinarily neglected.

Engineers, in practice, frequently assume an equivalent length of straight pipe to allow for the loss occasioned by elbows and globe valves. The assumption that is frequently made is to add to the measured length of line a length equal to 40 diameters of the pipe for each 90 degree elbow and 60 diameters for each globe valve.

For further data on the loss through fittings, etc., and the allowable velocity of water through pipes see the chapter on "Pumps," Volume II.

Example. A 2-inch pipe 300 ft. long with five–90° elbows and two globe valves is to carry 60 gallons per min. Required the pressure loss in the line.

From the chart Fig. 6 we find that the velocity will be approximately 6 ft. per sec. and that the friction loss in the straight run of pipe will be 3 × 3.25 = 9.75 lb. per sq. in. This is equivalent to a head of 9.75 × 2.3 or 22.4 feet.

The loss through 5 elbows is $5 \times 0.984 \times \frac{6^2}{2g} = 2.75$ ft.

The loss through 2 globe valves is $2 \times 1\frac{1}{2} \times 0.984 \times \frac{6^2}{2g} = 1.65$ ft.

The loss of head at entrance is $0.50 \times \frac{6^2}{2g} = 0.28$ ft.

The total estimated loss of head is therefore 22.4 + 2.75 + 1.65 + 0.28 = 27.08 ft.

Measurement of the Flow of Water. The weight of the liquid delivered in a unit of time may be determined either directly or indirectly. To determine the weight delivered directly, it is necessary to use weighing tanks and scales or to measure the volume delivered in a tank of known dimensions. In the latter case the density of the liquid, by which the volume is multiplied to obtain the weight, must be known. Owing to the large size of tanks necessary when the quantity discharged is considerable direct measurement is frequently impractical. The indirect methods of determining the weight of liquid delivered depend upon the use of weirs, orifices, meters, Pitot tube and the Venturi tube.

The V-Notch Weir. The apparatus consists of a tank divided into two chambers by a dividing sheet as shown by Fig. 7. A 90° V-notch weir is inserted in the top of the dividing sheet.

Behind the weir is the so-called surge chamber or tumbling bay. The tumbling bay is provided with a hook gage with scale and vernier as shown. The reading on the scale is noted when the point of the hook is on the level with the bottom of the V-notch. A reading is made, after the

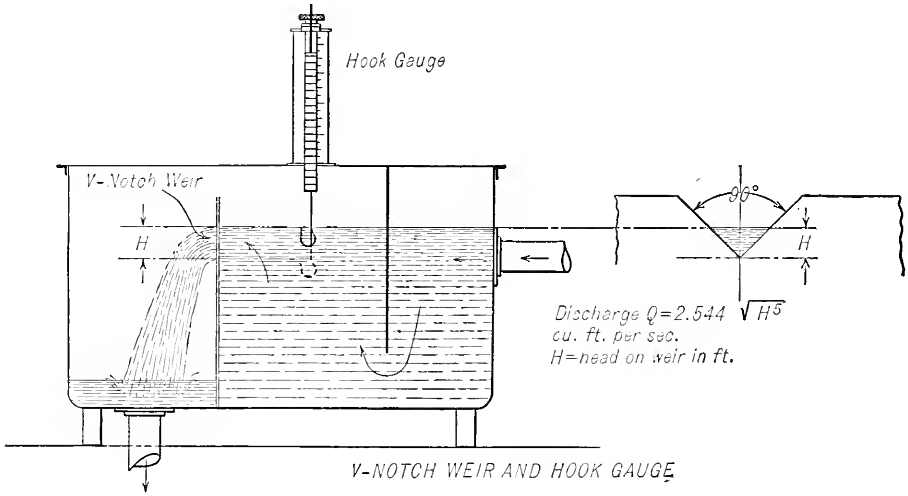


Fig. 7.

flow starts, by raising the gage until the point of the hook begins to pierce the surface of the water. The difference between the two readings gives the head producing the flow over the weir. The formula for the 90° V-notch weir as stated by *Professor James Thompson* is:

$$Q = 2.544 \sqrt{H^5}$$

in which

Q = volume flowing, cu. ft. per sec.
H = head on the weir in ft.

Where possible to adopt in practice, the V-notch weir will give consistent results and is quite extensively used in connection with the open type of feed-water heater. A recording device is readily attached to this apparatus through the medium of a float placed in a well which is in communication with the tumbling bay.

The Venturi Tube. For the measurement of flow in pipes under pressure the Venturi tube (Fig. 8) is a reliable form of meter and is extensively used in practice where accurate and consistent results are desired.

The head or pressure difference H between A in the "up-stream" portion of the contracted tube and B at the throat is made use of in determining the velocity at the throat.

$$V = \frac{A_a}{\sqrt{A^2_a - A^2_b}} \sqrt{2gH} \quad \ldots \ldots \quad (1)$$

in which

V = velocity at the throat, ft. per sec.
A_a = area of "up-stream" section of tube, sq. ft.
A_b = area of "throat" section, sq. ft.
H = difference in head measured in ft. of water column by the manometer.

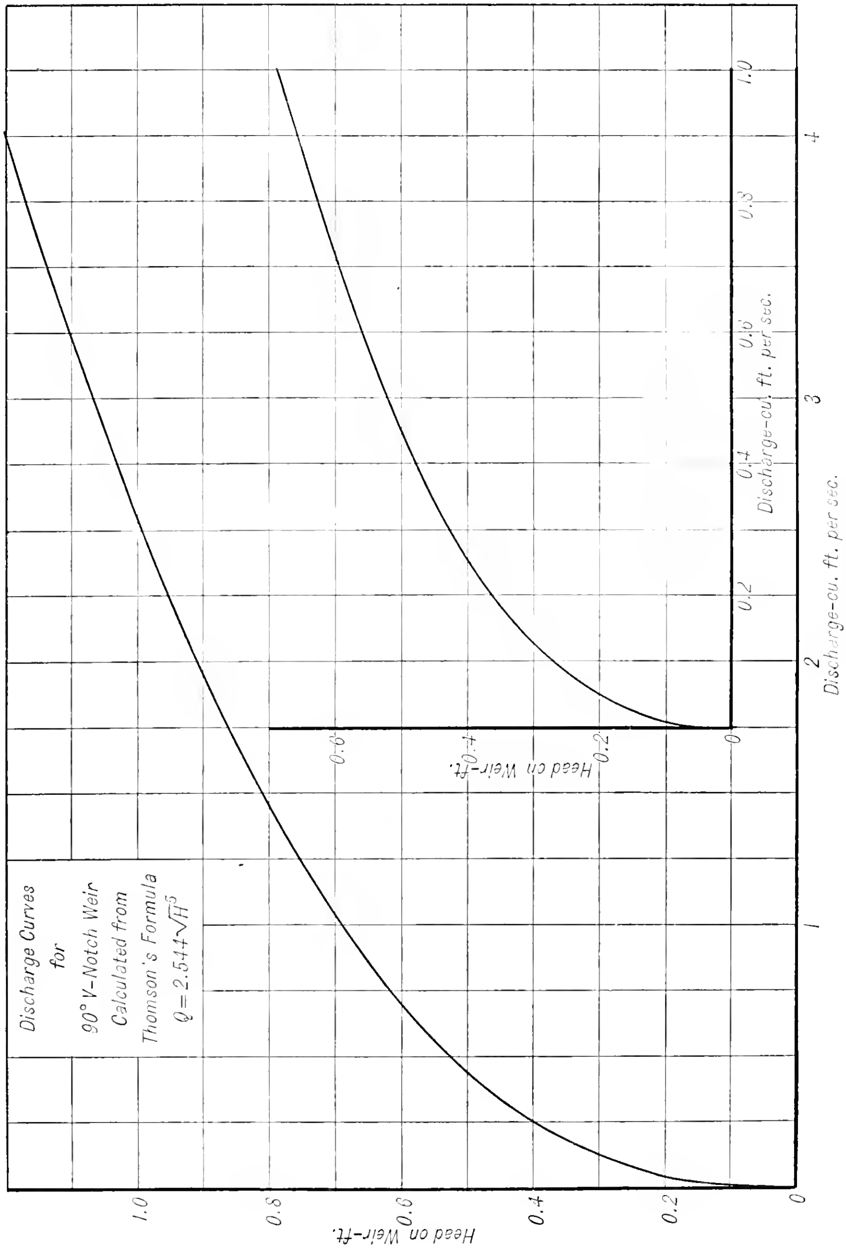


FIG. 7A. DISCHARGE CAPACITIES FOR 90° V-NOTCH WEIR.

The quantity of water discharged is,

$Q \quad = A_b\ V$ cu. ft. per sec. . . . . (2)

The velocity as determined by the above formula gives results within 3 per cent of the correct value. For extreme accuracy the meter should be calibrated by actually weighing the water for different rates of flow.

Measurement of Flow by Means of the Pitot Tube. As previously shown the Pitot tube

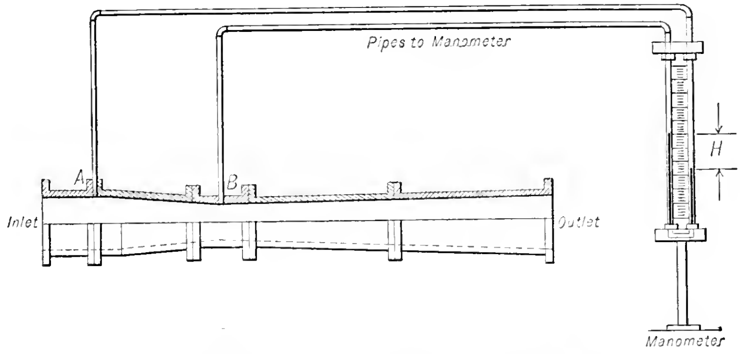


FIG. 8. VENTURI TUBE WITH INDICATING MANOMETER.

indicates the *total pressure* at the point of measurement. If a Pitot tube be placed at the discharge end of a pipe the reading obtained is the velocity head at the center of the pipe.

It is a well-known fact that the velocity is greatest at the center of the pipe and least at the walls. The ratio between these velocities being approximately two to one, for accurate work a traverse of the pipe should be made, as described in the chapter on "Hot Blast Heating," and the relation between the velocity at the center and the mean velocity established.

The traverse velocity curve approximates quite closely an ellipse. The mean average velocity is very nearly equal to 0.84 × the velocity as determined from the reading taken at the center of the pipe.

Let h_v = the velocity head measured at the center of the pipe in feet of water.

V = velocity at center of pipe in ft. per sec.

V_m = mean average velocity, ft. per sec.

Then $V_m = 0.84\ V = 0.84\sqrt{2gh_v}$

STEAM

Properties of Steam. Steam is water vapor, which exists in the vaporous condition due to the fact that sufficient heat has been added to the water, from which the steam has been formed, to supply the latent heat of evaporation, and change the liquid into vapor. This change in state takes place at a definite and constant temperature, which is determined solely by the pressure of the steam. A change in pressure will always be accompanied by a change in the temperature at which ebullition or boiling will occur, and there will be a corresponding change in the latent heat.

The properties of steam, together with other characteristics, are tabulated in the steam tables. See Table 5.

Steam in contact with the water from which it has been generated is known as *saturated steam,* and may be known as *dry saturated steam,* or as *wet saturated steam.* The latter contains more or less actual water in the form of mist or "priming" as it is called.

If dry, saturated steam be heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated,* that is, its temperature will be higher than that of saturated steam at the same pressure.

A conception of the relation between the properties or characteristics of steam, and the manner in which the changes in state, temperature and pressure are brought about is shown in Fig. 12 and described in the following paragraphs.

Generation of Steam. Consider a frictionless cylinder, Fig. 9, containing 1 lb. of water at 32° F. Also consider the pressure of the atmosphere to be 14.7 lb. per sq. in. and to be replaced by that of the piston *B*. When heat is applied to the cylinder the temperature of the water

FIG. 9.

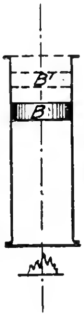


FIG. 10.

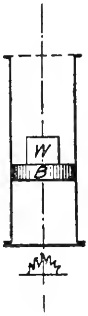


FIG. 11.

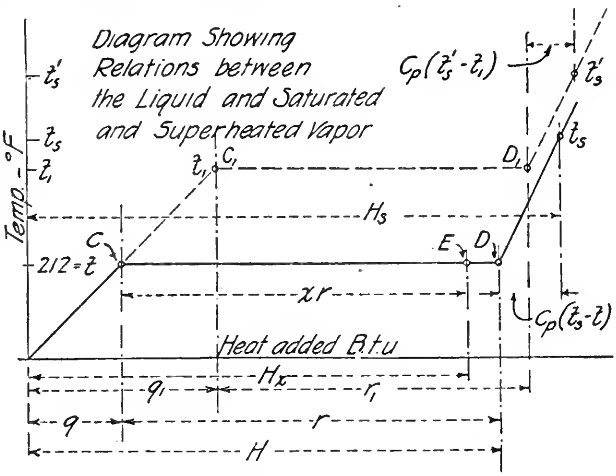


FIG. 12.

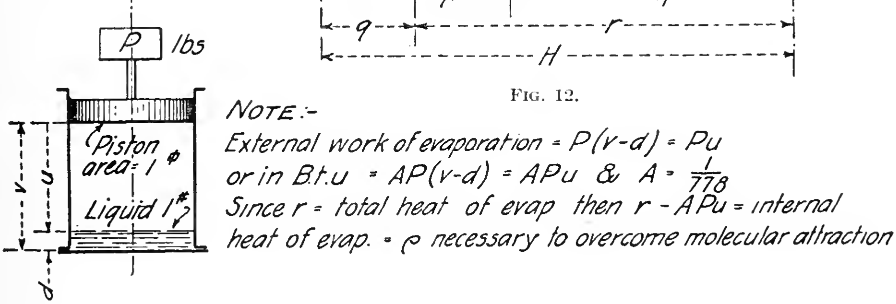


FIG. 13.

rises until the boiling point, 212° F., is reached. The heat necessary to raise the temperature from 32° F. to the boiling point is known as the "heat of the liquid" or "sensible heat," and is denoted by the symbol q. This condition is denoted on Fig. 12 by the point C. The average specific heat of water between 32° F. and 212° F. is 1, hence the number of British thermal units (B.t.u.) necessary to raise the temperature of the water this amount is 212 − 32 or 180 B.t.u.

When more heat is added the water begins to evaporate and expand at constant temperature until, as in Fig. 10, the water is entirely changed into steam. This condition is also shown on Fig. 12, by the point D. The heat thus added is known as the "latent heat of evaporation" and is denoted by the symbol r. This heat r is subdivided into two parts. See Fig. 13. First the attraction between the molecules must be broken down. This is known as the internal latent heat and is denoted by the symbol ρ; next the external resistance must be overcome, the weight P being raised against gravity as in Fig. 13. The heat thus added is known as external latent heat and is designated by the symbols APu, where u is the change in volume, in cu. ft. of one pound of water, A is 1/778 and P is the pressure of the atmosphere in pounds per sq. ft. (barometric pressure). It is evident then that the latent heat $r = \rho + APu$, or $\rho = r - APu$. The term APu is the heat equivalent of the work performed for the change in volume from water to steam.

The heat added from the starting point (32° F.) is known as *total heat* (H) or $q + r = H$. If more heat is added, the pressure remaining constant, the temperature of the steam rises and the steam becomes what is known as *superheated steam*. The heat added is equal to the mean specific heat (C_p) of the steam times the change in temperature ($t_s - 212$). Specific heat of steam is the B.t.u. or heat required to raise the temperature of 1 pound of the steam 1° F. Since the specific heat of steam is less than that of water, the slope of this line becomes greater than that of the water line. The point is now located at t_s on Fig. 12, and the steam has increased in volume in the cylinder of Fig. 10 until the piston occupies the dotted position B'.

If instead of the above condition of pressure, additional pressure be added as shown by the weight W in Fig. 11, the temperature of the boiling point will be raised from the temperature of 212° F. to some other point as t_1 in Fig. 12. As may be seen by this figure, the sensible heat q has been increased to q_1. When more heat is added the water is evaporated at the temperature t_1 and if heat again be added the saturated steam will become superheated.

Quality. The proportion of the dry steam per pound of steam delivered by the boiler is known as the quality of the steam and is represented by the symbol x, and the heat (H_x) contained in the steam above 32° F. is $q + xr$ and the state point is located at E in Fig. 12.

The volume of a pound of steam is known as the *specific volume* (v), and, as may be seen by comparing Figs. 10 and 11, decreases as the pressure increases. The reciprocal of this or weight of steam per cu. ft. is known as the *density* and is denoted by d or $\frac{1}{v}$.

The relation between pressure and specific volume for dry saturated steam is given by the experimental equation (*Goodenough*) as $pv^{1 \cdot 0631} = 484.2$ in which

p = pressure in pounds per sq. in.

v = specific volume.

Another quantity known as *entropy* is made use of in calculations relating to steam engines and turbines, and is defined as the ratio obtained by dividing the quantity of heat added to a substance by the absolute temperature at which it is added. The *entropy of the liquid* is represented by s' or n, the *entropy of vaporization* by $\frac{r}{T}$ and the *entropy of the vapor* s'' or s. The use of entropy is explained under the "Rankine Cycle," in the chapter on "Steam Engines."

The total heat (H) of a dry saturated vapor for any pressure and temperature is the sum of the heats required to raise the temperature of one pound of the liquid from the freezing point to the given temperature and corresponding pressure and *entirely* vaporize it at this pressure. For this case $x = 1$, consequently $H = (\rho + APu) + q = r + q$; $H = 1151.7 + 0.3745\ (t - 212) - 0.00055\ (t - 212)^2$, as stated by *Marks and Davis*.

The total heat (H_x) of wet vapor at any pressure and temperature is the sum of the heats required to raise the temperature of one pound of the liquid from the freezing point to the given temperature and corresponding pressure and to vaporize the part x at this pressure. For this case,

$$H_x = xr + q.$$

It is manifestly incorrect to say this is the heat *in* the vapor as the APu is not heat in the vapor, but the external work performed by the vapor while evaporating.

Heat Content of Saturated Steam. This by definition is $i'' = q + \rho + APv''$ in which v'' is the specific volume of the steam.

The total heat of saturated steam by definition is $H = q + \rho + AP\ (v'' - v')$, in which v' is the specific volume of the liquid.

As v' is small compared with v'' the term $A\rho v'$ may be neglected, except at very high temperatures and pressures, and i'' and H may be considered equal.

In recent steam tables the values of i'' instead of H are usually tabulated.

Superheated Steam or Vapor. Superheated steam is defined as water vapor which has been

heated, out of contact with its liquid, until its temperature is higher than that of saturated vapor at the same pressure. Moreover, if the temperature or degree of superheat is far removed from the temperature of saturation the superheated vapor will follow the laws of perfect gases quite closely, ($PV = MRT$), except at high pressures or low temperatures. See "Air and Other Gases."

The relation between pressure, volume, and temperature, experimentally determined for superheated steam is $V + 0.256 = 0.5962 \frac{T}{p}$ which *Linde* gives as a rough approximation, where

V = specific volume.
T = absolute temperature.
p = pounds per sq. in.

The *specific heat of superheated steam* is not constant as shown by the experiments of *Knoblauch* and *Jakob*, and others. Curves of mean specific heat are shown in Fig. 14. For any degree of superheat the mean specific heat between the saturation state and the given state is given by the ordinate corresponding to the given degree of superheat and the given pressure. For example, at a pressure of 150 lb. per sq. in. absolute the mean specific heat for 240° superheat is 0.529.

The heat content of superheated steam or vapor may be expressed by the equation $H_s = q + r + C_p (t_s - t) = H + C_p (t_s - t)$ where t_s = temperature of superheated vapor and t = temperature of saturated vapor at the corresponding pressure, q = heat of the liquid at t, and r = heat of vaporization at temperature t. C_p = mean specific heat of superheated vapor, H = total heat of one pound of dry saturated steam, and H_s = total heat of one pound of superheated steam.

Throttling Calorimeter. The expressions for heat content of a liquid and its vapor, and the heat content of superheated steam, are made use of in finding the part x of a mixture that exists as wet vapor, within certain limits.

This is commonly known as the determination of moisture or "priming" in steam by means of the "throttling" or superheating calorimeter and the necessary data applicable to the above

TABLE 5

PROPERTIES OF SATURATED STEAM

(G. A. Goodenough)

Pressure		Temp., ° F.	Volume, Cu. Ft. per Lb.	Weight, Lb. per Cu. Ft.	Heat Content in B.t.u.		Latent Heat in B.t.u.		Entropy		
In. of Mercury	Lb. per Sq. In.				of Liquid	of Vapor	of Vaporization	Internal	of Liquid	of Vaporization	of Vapor
p	—	t	v''	$1/v''$	i'	i''	r	ρ	s'	r/T	s''
				d	q	H			n		s
2.036	1	101.76	333.3	0.00300	69.76	1105.4	1035.6	973.9	0.1327	1.8448	1.9775
4.072	2	126.10	173.6	.00576	94.02	1116.2	1022.2	957.9	.1750	1.7452	1.9203
4.6	2.260	130.64	154.8	.00646	98.55	1118.2	1019.7	954.9	.1827	1.7275	1.9103
4.8	2.358	132.24	148.8	.00672	100.14	1118.9	1018.8	953.8	.1854	1.7214	1.9068
6.108	3	141.49	118.7	.00843	109.38	1122.9	1013.5	947.6	.2009	1.6862	1.8871
8.144	4	152.99	90.6	.01104	120.9	1127.9	1007.0	939.9	.2199	1.6438	1.8637
10.180	5	162.25	73.5	.01360	130.1	1131.7	1001.6	933.6	.2348	1.6107	1.8456
12.216	6	170.07	62.0	.01614	137.9	1135.0	997.1	928.2	.2473	1.5835	1.8308
14.25	7	176.85	53.7	.01864	144.7	1137.8	993.1	923.6	.2581	1.5603	1.8184
16.29	8	182.87	47.35	.02112	150.8	1140.3	989.5	919.4	.2675	1.5402	1.8077
18.32	9	188.28	42.41	.02358	156.2	1142.5	986.3	915.6	.2759	1.5223	1.7982
20.36	10	193.21	38.43	.02602	161.1	1144.4	983.3	912.2	.2835	1.5062	1.7897
22.40	11	197.75	35.16	.02844	165.7	1146.2	980.5	909.0	.2905	1.4916	1.7821
24.43	12	201.96	32.41	.03086	169.9	1147.9	978.0	906.0	.2969	1.4783	1.7752
26.47	13	205.88	30.07	.03326	173.8	1149.4	975.6	903.2	.3028	1.4659	1.7687
28.50	14	209.56	28.06	.03564	177.5	1150.8	973.3	900.6	.3083	1.4545	1.7628
29.92	14.697	212	26.81	.03730	180.0	1151.7	971.7	898.8	.3120	1.4469	1.7589
30	14.74	212.13	26.75	.03739	180.1	1151.8	971.7	898.8	.3122	1.4465	1.7587

TABLE 5—(*Continued*)

PROPERTIES OF SATURATED STEAM

(*G. A. Goodenough*)

Absolute Pressure, Lb. per Sq. In.	Temp., ° F.	Volume, Cu. Ft. per Lb.	Weight, Lb. per Cu. Ft.	Heat Content in B.t.u.		Latent Heat in B.t.u.		Entropy		
				of Liquid	of Vapor	of Vaporization	Internal	of Liquid	of Vaporization	of Vapor
p	t	v''	$1/v''$	i'	i''	r	ρ	s'	r/T	s''
			d	q	H			n		s
16	216.3	24.76	0.04038	184.3	1153.4	969.1	895.8	0.3184	1.4337	1.7521
18	222.4	22.18	.04508	190.5	1155.7	965.2	891.4	.3274	1.4153	1.7427
20	228.0	20.10	.04976	196.0	1157.7	961.7	887.3	.3356	1.3987	1.7343
22	233.1	18.38	.0544	201.2	1159.6	958.4	883.6	.3430	1.3837	1.7267
24	237.8	16.95	.0590	206.0	1161.3	955.3	880.1	.3499	1.3698	1.7197
26	242.2	15.73	.0636	210.4	1162.8	952.4	876.8	.3563	1.3570	1.7133
28	246.4	14.67	.0681	214.6	1164.3	949.7	873.7	.3622	1.3452	1.7074
30	250.3	13.76	.0727	218.6	1165.7	947.1	870.7	.3679	1.3340	1.7019
32	254.0	12.95	.0772	222.4	1166.9	944.6	867.9	.3731	1.3236	1.6967
34	257.6	12.24	.0818	225.9	1168.1	942.2	865.2	.3781	1.3137	1.6918
36	260.9	11.60	.0862	229.4	1169.2	939.9	862.7	.3829	1.3044	1.6873
38	264.2	11.03	.0907	232.6	1170.3	937.7	860.2	.3874	1.2956	1.6830
40	267.2	10.51	.0951	235.8	1171.3	935.5	857.8	.3917	1.2871	1.6788
42	270.2	10.04	.0996	238.8	1172.2	933.5	855.5	.3958	1.2791	1.6749
44	273.0	9.61	.1040	241.7	1173.2	931.5	853.3	.3998	1.2714	1.6712
46	275.8	9.22	.1085	244.5	1174.0	929.6	851.2	.4036	1.2640	1.6676
48	278.4	8.86	.1129	247.2	1174.8	927.7	849.1	.4072	1.2570	1.6642
50	281.0	8.53	.1173	249.8	1175.6	925.9	847.1	.4108	1.2501	1.6609
52	283.5	8.22	.1217	252.3	1176.4	924.1	845.1	.4142	1.2436	1.6577
54	285.9	7.93	.1261	254.7	1177.1	922.4	843.2	.4174	1.2373	1.6547
56	288.2	7.67	.1304	257.1	1177.8	920.7	841.4	.4206	1.2311	1.6517
58	290.5	7.42	.1348	259.5	1178.5	919.0	839.5	.4237	1.2252	1.6489
60	292.7	7.18	.1392	261.7	1179.1	917.4	837.8	.4267	1.2195	1.6462
62	294.9	6.97	.1435	263.9	1179.7	915.8	836.0	.4296	1.2139	1.6435
64	296.9	6.76	.1479	266.1	1180.3	914.3	834.3	.4324	1.2085	1.6409
66	299.0	6.57	.1522	268.2	1180.9	912.7	832.7	.4352	1.2032	1.6384
68	301.0	6.39	.1566	270.2	1181.5	911.2	831.1	.4379	1.1981	1.6360
70	302.9	6.22	.1609	272.2	1182.0	909.8	829.5	.4405	1.1931	1.6336
72	304.8	6.05	.1652	274.2	1182.5	908.3	827.9	.4431	1.1883	1.6313
74	306.7	5.90	.1695	276.1	1183.0	906.9	826.4	.4456	1.1835	1.6291
76	308.5	5.75	.1738	278.0	1183.5	905.5	824.9	.4480	1.1789	1.6269
78	310.3	5.61	.1781	279.8	1184.0	904.2	823.4	.4504	1.1744	1.6248
80	312.0	5.48	.1824	281.6	1184.4	902.8	821.9	.4527	1.1700	1.6227
82	313.7	5.35	.1868	283.4	1184.9	901.5	820.5	.4550	1.1657	1.6207
84	315.4	5.23	.1910	285.1	1185.3	900.2	819.1	.4572	1.1615	1.6187
86	317.1	5.12	.1953	286.8	1185.7	898.9	817.7	.4594	1.1574	1.6168
88	318.7	5.01	.1996	288.5	1186.1	897.7	816.3	.4615	1.1534	1.6149
90	320.3	4.905	.2039	290.1	1186.5	896.4	815.0	.4636	1.1495	1.6131
92	321.8	4.805	.2081	291.7	1186.9	895.2	813.7	.4657	1.1456	1.6113
94	323.3	4.709	.2124	293.3	1187.3	894.0	812.4	.4677	1.1419	1.6096
96	324.8	4.617	.2166	294.8	1187.7	892.8	811.1	.4697	1.1381	1.6079
98	326.3	4.528	.2209	296.4	1188.0	891.6	809.8	.4717	1.1345	1.6062
100	327.8	4.442	.2251	297.9	1188.4	890.5	808.6	.4736	1.1309	1.6045
102	329.2	4.359	.2294	299.4	1188.7	889.3	807.4	.4755	1.1274	1.6028
104	330.7	4.279	.2337	300.9	1189.0	888.2	806.1	.4773	1.1239	1.6012
106	332.0	4.202	.2380	302.3	1189.4	887.1	804.9	.4791	1.1205	1.5996
108	333.4	4.128	.2422	303.7	1189.7	885.9	803.8	.4809	1.1172	1.5981
110	334.8	4.057	.2465	305.1	1190.0	884.8	802.6	.4827	1.1138	1.5965
112	336.1	3.988	.2508	306.5	1190.3	883.7	801.4	.4844	1.1106	1.5950
114	337.4	3.921	.2550	307.9	1190.6	882.7	800.3	.4861	1.1074	1.5935
116	338.7	3.857	.2593	309.2	1190.8	881.6	799.2	.4878	1.1043	1.5921
118	340.0	3.795	.2635	310.6	1191.1	880.6	798.0	.4895	1.1012	1.5907
120	341.3	3.735	.2678	311.9	1191.4	879.5	796.9	.4911	1.0982	1.5893
122	342.5	3.676	.2720	313.2	1191.6	878.5	795.8	.4927	1.0952	1.5879
124	343.7	3.620	.2762	314.4	1191.9	877.5	794.8	.4943	1.0922	1.5865
126	345.0	3.566	.2805	315.7	1192.1	876.4	793.7	.4958	1.0894	1.5852
128	346.2	3.513	.2847	316.9	1192.4	875.4	792.6	.4974	1.0865	1.5838
130	347.4	3.461	.2889	318.2	1192.6	874.4	791.6	.4989	1.0836	1.5825
132	348.5	3.412	.2931	319.4	1192.9	873.5	790.5	.5004	1.0808	1.5812
134	349.7	3.363	.2973	320.6	1193.1	872.5	789.5	.5019	1.0781	1.5800
136	350.8	3.316	.3016	321.8	1193.3	871.5	788.5	.5033	1.0754	1.5787
138	352.0	3.270	.3058	323.0	1193.5	870.5	787.4	.5048	1.0727	1.5775
140	353.1	3.226	.3100	324.2	1193.7	869.6	786.4	.5062	1.0700	1.5762
142	354.2	3.182	.3142	325.3	1193.9	868.6	785.4	.5076	1.0674	1.5750
144	355.3	3.140	.3184	326.5	1194.1	867.7	784.5	.5090	1.0648	1.5738

TABLE 5—(*Continued*)

PROPERTIES OF SATURATED STEAM

(*G. A. Goodenough*)

Absolute Pressure, Lb. per Sq. In.	Temp., ° F.	Volume, Cu. Ft. per Lb.	Weight Lb. per Cu. Ft.	Heat Content in B.t.u.		Latent Heat in B.t.u.		Entropy		
				of Liquid	of Vapor	of Vaporization	Internal	of Liquid	of Vaporization	of Vapor
p	t	v''	$1/v''$	i'	i''	r	ρ	s'	r/T	s''
			d	q	H			n		s
146	356.3	3.099	0.3227	327.6	1194.3	866.8	783.5	0.5104	1.0623	1.5727
148	357.4	3.059	.3269	328.7	1194.5	865.8	782.5	.5117	1.0598	1.5715
150	358.5	3.020	.3311	329.8	1194.7	864.9	781.6	.5131	1.0573	1.5704
152	359.5	2.982	.3353	330.9	1194.9	864.0	780.6	.5144	1.0548	1.5692
154	360.5	2.945	.3396	332.0	1195.1	863.1	779.7	.5157	1.0524	1.5681
156	361.6	2.909	.3438	333.1	1195.3	862.3	778.7	.5170	1.0500	1.5670
158	362.6	2.874	.3480	334.1	1195.5	861.4	777.8	.5183	1.0476	1.5659
160	363.6	2.839	.3522	335.2	1195.7	860.5	776.9	.5196	1.0453	1.5649
162	364.6	2.806	.3564	336.2	1195.8	859.6	776.0	.5209	1.0429	1.5638
164	365.6	2.773	.3606	337.3	1196.0	858.7	775.1	.5221	1.0406	1.5627
166	366.5	2.741	.3648	338.3	1196.2	857.9	774.2	.5233	1.0384	1.5617
168	367.5	2.710	.3691	339.3	1196.3	857.0	773.3	.5245	1.0361	1.5607
170	368.5	2.679	.3733	340.3	1196.5	856.2	772.4	.5258	1.0339	1.5597
172	369.4	2.649	.3775	341.3	1196.6	855.3	771.5	.5270	1.0317	1.5587
174	370.4	2.620	.3817	342.3	1196.8	854.5	770.6	.5281	1.0295	1.5577
176	371.3	2.591	.3859	343.3	1196.9	853.6	769.8	.5293	1.0274	1.5567
178	372.2	2.563	.3901	344.3	1197.1	852.8	768.9	.5305	1.0252	1.5557
180	373.1	2.536	.3943	345.2	1197.2	852.0	768.0	.5316	1.0231	1.5547
182	374.0	2.509	.3985	346.2	1197.4	851.2	767.2	.5328	1.0210	1.5538
184	374.9	2.483	.4027	347.1	1197.5	850.4	766.4	.5339	1.0189	1.5528
186	375.8	2.457	.4069	348.1	1197.6	849.5	765.5	.5350	1.0169	1.5519
188	376.7	2.432	.4111	349.0	1197.8	848.7	764.7	.5361	1.0148	1.5509
190	377.6	2.408	.4154	350.0	1197.9	847.9	763.9	.5372	1.0128	1.5500
192	378.5	2.383	.4196	350.9	1198.0	847.1	763.0	.5383	1.0108	1.5491
194	379.3	2.360	.4238	351.8	1198.1	846.3	762.2	.5394	1.0089	1.5482
196	380.2	2.337	.4280	352.7	1198.2	845.6	761.4	.5404	1.0069	1.5473
198	381.0	2.314	.4322	353.6	1198.4	844.8	760.6	.5415	1.0049	1.5464
200	381.9	2.292	.4364	354.5	1198.5	844.0	759.8	.5426	1.0030	1.5456
202	382.7	2.270	.4406	355.4	1198.6	843.2	759.0	.5436	1.0011	1.5447
204	383.5	2.248	.4448	356.2	1198.7	842.5	758.2	.5446	0.9992	1.5438
206	384.4	2.227	.4490	357.1	1198.8	841.7	757.4	.5457	.9973	1.5430
208	385.2	2.206	.4532	358.0	1198.9	840.9	756.7	.5467	.9954	1.5421
210	386.0	2.186	.457	358.8	1199.0	840.2	755.9	.5477	.9936	1.5413
212	386.8	2.166	.462	359.7	1199.1	839.4	755.1	.5487	.9918	1.5405
214	387.6	2.147	.466	360.5	1199.2	838.7	754.3	.5497	.9900	1.5396
216	388.4	2.128	.470	361.4	1199.3	837.9	753.6	.5507	.9881	1.5388
218	389.2	2.109	.474	362.2	1199.4	837.2	752.8	.5516	.9864	1.5380
220	390.0	2.090	.478	363.0	1199.5	836.5	752.1	.5526	.9846	1.5372

equations for heat content are obtained by causing a sample of the wet saturated steam from the boiler, at high pressure, to pass through a small orifice and expand to a lower pressure without doing any external work. The total heat of saturated steam at the lower pressure is less than the total heat of saturated steam at the higher pressure. Since, however, the steam in expanding from a high pressure to a lower pressure does no external work, and assuming no radiation loss, no heat is added or taken away from the system. Then according to the Law of Conservation of Energy, since the velocity before and after expansion is zero, it may readily be shown that the heat content per pound must also be the same before and after expansion and some heat will be available for drying and superheating the steam at the lower pressure.

Example. Refer to Fig. 15, showing a *throttling calorimeter* connected to a steam pipe, and assume that steam at 160 lb. gage is flowing in the steam pipe. Some of this steam enters the holes in the "sampling pipe" if the gate valve is opened wide, and passes by the upper thermometer, which records its temperature of 370.7° F., and then through a ⅛" diameter orifice in the disk between the two flanges, where free expansion takes place; the pressure changing from 160 lb. gage to that of the atmosphere. The lower thermometer indicates the temperature after expansion, and if no superheating takes place it will read 212° F. at normal atmospheric pressure.

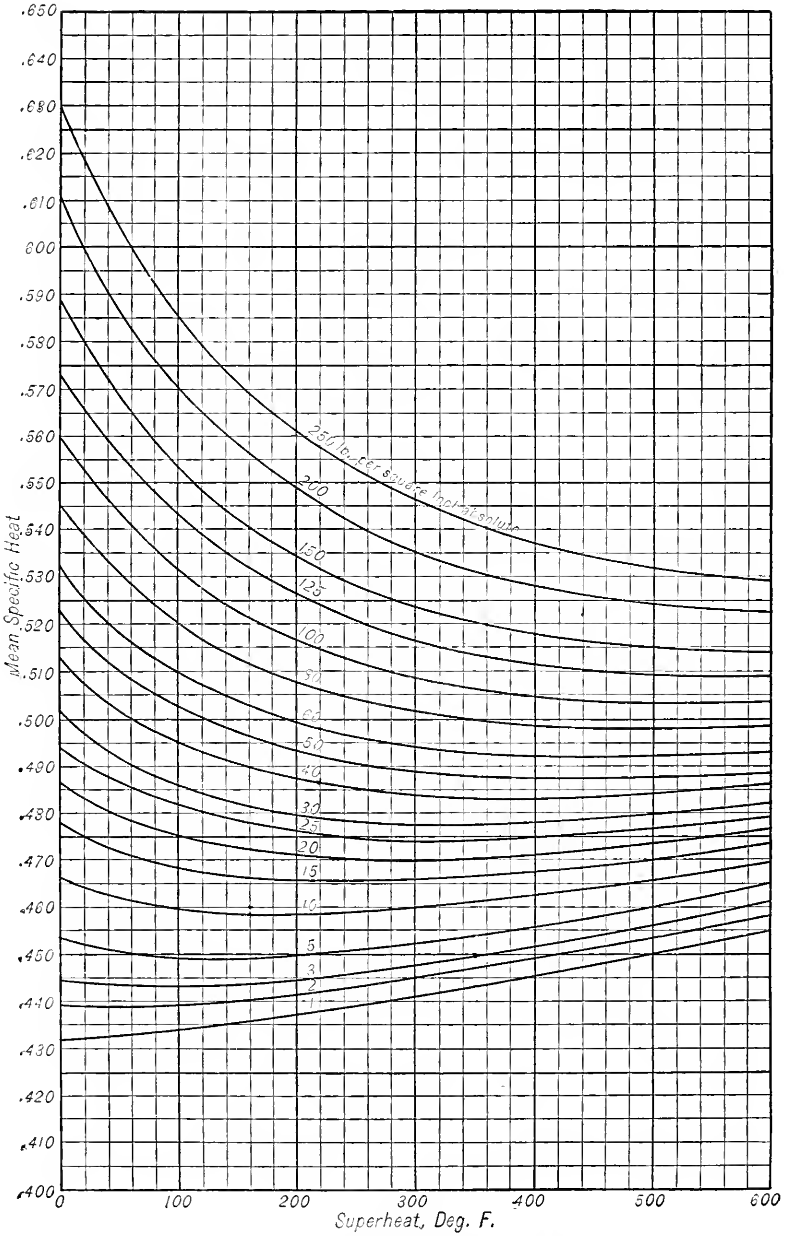


Fig. 14. Mean Specific Heat Curves.

(G. A. Goodenough)

If the steam is absolutely dry and saturated in the main steam pipe the total heat H is 1196.9 B.t.u. per pound, and at atmospheric pressure the total heat H per pound of dry saturated steam is 1151.7 B.t.u. As the heat content must be the same after free expansion as before there is available 1196.9 − 1151.7 or 45.2 B.t.u., which goes to superheat the steam at the lower pressure. The amount of superheat or the number of degrees above the saturation temperature, corresponding to atmospheric pressure, to which the steam after free expansion will be raised is $\frac{45.2}{0.47} = 96.2°$, where 0.47 is the mean specific heat of superheated steam at atmospheric pressure. Hence the lower thermometer will read 212 + 96.2 = 308.2° F., if no moisture is present in the original steam.

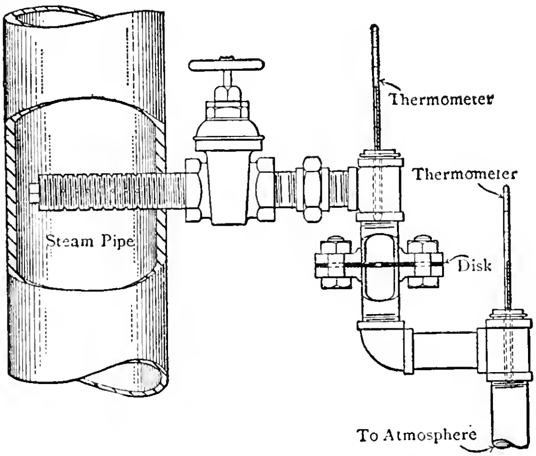


FIG. 15. THROTTLING CALORIMETER AND SAMPLING NOZZLE.

If the original steam contains, say 1 per cent of moisture, it will take 8.5 B.t.u. to evaporate this moisture at 370.7° F. since the latent heat at this temperature is 854.2 B.t.u. per lb. We will then have left for superheating 45.2 − 8.5 = 36.7 B.t.u. or the steam will be superheated only $\frac{36.7}{0.47} = 78.1°$ F.

It is readily seen that as the moisture increases less and less heat will be available for superheating, until finally no superheating will occur and *the limit of moisture determination by the throttling calorimeter* for steam at this pressure will have been reached.

The general formula for finding the quality of steam by this apparatus at any pressure is given below:

$H_x = H_s$ where $H_x = x\,r_1 + q_1$ = total heat of one pound of steam at the initial pressure.

$H_s = r_2 + q_2 + C_p\,(t_s - t_2)$ = total heat of one pound of steam at the final or atmospheric pressure.

$$\text{Hence } x\,r_1 + q_1 = r_2 + q_2 + C_p\,(t_s - t_2)$$

$$x = \frac{r_2 + q_2 + C_p\,(t_s - t_2) - q_1}{r_1}$$

r_1 and r_2 = latent heat of vaporization at the initial and final pressures respectively.
q_1 and q_2 = heat of the liquid at the initial and final pressures respectively.
C_p = mean specific heat of superheated steam (see Fig. 14).
t_s = temperature of steam after superheating.
t_2 = temperature of saturated steam at the final pressure (atmosphere).

The limit of moisture or maximum value of $1 - x$, is found by making $t_s = t_2$ for any given case and solving for x. These limits range from 2.88 per cent at 50 lb. gage to 7.17 per cent moisture at 250 lb. gage, at sea level.

Practically there are slight errors in the process due to the exposed stem of the thermometer, and the radiation loss from the instrument. The stem correction can be made as already indicated, and by heavily lagging the instrument the radiation loss can be largely overcome. Both errors tend to reduce the reading of the lower thermometer, t_2.

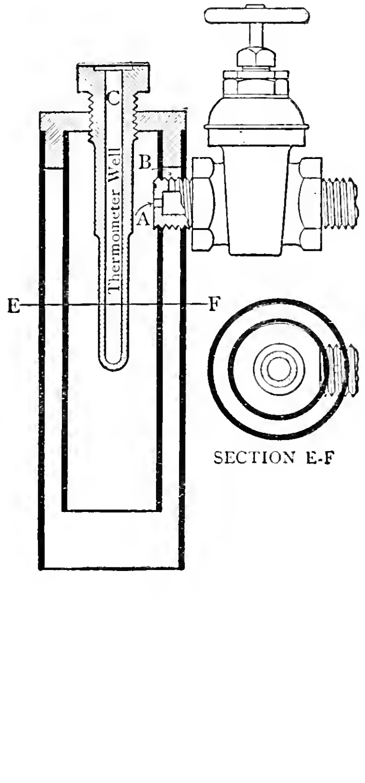


Fig. 16. Compact Throttling Calorimeter.

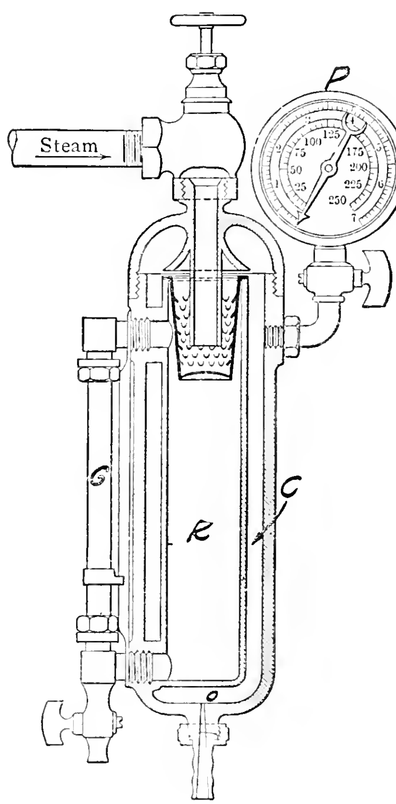


Fig. 17. Separating Calorimeter.

A very compact form of the throttling calorimeter is shown in Fig. 16.

For very "wet" steam *a separating calorimeter* must be used, and a section of such an apparatus is shown in Fig. 17. This apparatus is in effect a small separator which mechanically separates the entrained water from the steam and collects it in a reservoir (R) where its amount

is indicated in a gage glass (G), while dry steam only escapes at the orifice (O). This orifice is of known size, and if the pressure in the chamber (C) is known the weight of dry steam passing the orifice can be calculated, or a gage (P) can be calibrated to read directly, the weight of steam flowing in pounds, provided the absolute pressure is not less than 25.37 lb. where the orifice discharges into the atmosphere. For absolute pressures lower than this a calculation must be made as stated by the formula under "Flow of Steam through Orifices."

Mixtures of Air and Saturated Water Vapor. The method of calculating the weight of water vapor mixed with air, for various conditions of pressure and temperature, will be found under the chapter on "Air Conditioning." A table and diagram are included for convenience in solving problems relative to the subject.

Flow of Steam Through Pipes. Various formulas for the flow of steam through pipes have been advanced, having their basis upon *Bernoulli's* theorem of the flow of water through circular pipes with the proper modifications made for the variation in constants between steam and water. *Unwin's* formula based on *Weisbach's* work is very commonly used and may be stated as follows:

$$h = f \times \frac{2\,L}{D} \times \frac{v^2}{g} \text{ See "Friction Head due to Flow of Water"} \quad \ldots\ldots \quad (1)$$

in which h represents the loss of head in feet of the fluid flowing, in this case steam, which is passing with a velocity of v feet per second, through a pipe D feet in diameter, and L feet long; g represents the acceleration due to gravity, and f the coefficient of friction.

Numerous values have been given for this coefficient of friction, f, which, from experiment, apparently varies with both the diameter of pipe and the velocity of the passing steam. There are no authentic data on the rate of this variation with velocity, and, as in all experiments, the effect of change of velocity has seemed less than the unavoidable errors of observation, the coefficient is assumed to vary only with the size of the pipe.

Unwin established a relation for this coefficient for steam at a velocity of 100 feet per second.

$$f = \mathrm{K}\left(1 + \frac{3}{10\,D}\right) \quad \ldots\ldots \quad (2)$$

where K is a constant experimentally determined, and D the internal diameter of the pipe in feet.

If d represents the density of the steam or weight per cubic foot, and p the loss of pressure due to friction in pounds per square inch, then

$$p = \frac{hd}{144} \quad \ldots\ldots \quad (3)$$

and from equations (1), (2), and (3),

$$p = \frac{dv^2\,L}{72\,g\,D} \times K\left(1 + \frac{3}{10\,D}\right) \quad \ldots\ldots \quad (4)$$

To convert the velocity term into weight and to reduce to units ordinarily used let D_1 = the diameter of pipe in inches = $12D$, and w = the weight of steam in pounds per minute, then

$$w = 60v \times \frac{\pi}{4} \times \left(\frac{D_1}{12}\right)^2 \times d$$

$$\text{and, } v = \frac{9.6\,w}{\pi\,D_1^2\,d}$$

Substituting this value and that of D in formula (4)

$$p = 0.04839\,K\left(1 + \frac{3.6}{D_1}\right)\frac{w^2\,L}{d\,D_1^5} \quad \ldots\ldots \quad (5)$$

Some of the experimental determinations for the value of K for steam are:

$K = 0.0026$ (*R. C. Carpenter*).
$K = 0.0027$ (*G. H. Babcock*).

Substituting the value 0.0027 in formula (5) gives,

$$p = 0.000131\left(1 + \frac{3.6}{D_1}\right) \times \frac{w^2\,L}{d\,D_1^5} \qquad (6)$$

$$\text{and, } w = 87.5\left[\frac{p\,d\,D_1^5}{\left(1 + \frac{3.6}{D_1}\right) \times L}\right]^{\frac{1}{2}} \qquad (7)$$

in which the various symbols have already been defined.*

This formula is the one most generally accepted in this country for the flow of steam in pipes.

Equation (4) may be written,

$$V = 16{,}050\left[\frac{p\,D_1}{L\,d\left(1 + \frac{3.6}{D_1}\right)}\right]^{\frac{1}{2}}$$, in which V = velocity of the steam in ft. per min.

Equation (6) may be written,

$$p = A \times \frac{w^2\,L}{d} \qquad (8)$$

in which

$$A = \frac{0.000131\left(1 + \frac{3.6}{D_1}\right)}{D_1^5}$$

Equation (7) may be written,

$$w = C\left[\frac{p\,d}{L}\right]^{\frac{1}{2}} \qquad (9)$$

in which

$$C = 87.5\left[\frac{D_1^5}{\left(1 + \frac{3.6}{D_1}\right)}\right]^{\frac{1}{2}}$$

For values of A and C see Table 6.

Equivalent Length of Pipe for Each Globe Valve, Entrance, and Elbow. In addition to the loss of pressure due to friction, in straight pipe, there is also a loss of pressure due to a change in the velocity of the steam at the entrance to the pipe. This drop in pressure due to getting up velocity in the pipe is very slight and is seldom taken into account.

Elbows, globe valves, and a square-ended entrance to the pipe, such as occurs when steam is taken off through a tee at right angles to the main, all offer resistance to the flow of steam, thus causing a drop in pressure, which should be taken into account and proper allowance made for it.

Friction is greater through short radius elbows and tees, than through elbows and tees of long radius. The resistance offered by a globe valve is about ⅓ greater than that due to a short radius elbow, whereas gate valves offer practically no resistance to the flow, providing they are opened wide. The resistance offered by a square-ended opening, or at the outlet of a

*d, the density, is taken as the mean density at the initial and final pressures and in exact work on pipes up to 5" diameters actual internal diameters should be used.

tee where a branch is taken off at right angles, is about the same as that for a globe valve having the same size opening. The resistance offered by a long radius pipe bend is very slight and may be taken as equal to the resistance offered by the same length of straight pipe, or in other words, all pipe bends may be considered as straight pipe of equal length.

TABLE 6

Nominal Pipe Size—Inches	Actual Inside Diameter Inches = D_1	Values of Constant "C"	Values of Constant "A"	Equivalent Length of Pipe, in Feet, to be Added for each Globe Valve and Entrance	Equivalent Length of Pipe in Feet, to be Added for each 90° Elbow
1	1.047	46	0.000,466	2	1.5
1¼	1.38	102	.000,095	4	3.0
1½	1.61	159	.000,039	5	3.5
2	2.067	320	.000,009,5	7	5
2½	2.467	513	.000,003,6	10	6
3	3.066	977	.000,001,1	14	9
3½	3.548	1,410	.000,000,17	17	11
4	4.026	2,016	.000,000,24	20	13
4½	4.508	2,795	.000,000,13	24	16
5	5.045	3,724	.000,000,07	28	19
6	6.065	6,210	.000,000,026	37	24
7	7.023	9,198	.000,000,011	44	29
8	7.981	13,050	.000,000,005,8	53	35
9	8.937	17,787	.000,000,003,1	61	41
10	10.018	23,605	.000,000,001,8	70	47
11	11	30,276	.000,000,001,1	78	52
12	12	38,074	.000,000,000,7	86	58
	14	56,862	.000,000,000,31	106	70
	16	80,384	.000,000,000,15	123	82
	18	109,281	.000,000,000,083	143	95
	20	143,120	.000,000,000,048	162	107
	22	183,870	.000,000,000,03	181	120
	24	229,993	.000,000,000,02	200	132

It is customary to consider the resistance offered by valves and fittings, etc., as equivalent to a length of straight pipe which will offer the same resistance, or cause the same drop in pressure. When this equivalent length has been determined it should be added to length L in the formula, and p, or w, computed accordingly.

Equivalent length of straight pipe, in inches, to be added for each globe valve, or square-ended opening

$$= \frac{114\ D_1}{\left(1 + \frac{3.6}{D_1}\right)}.$$

Equivalent length of straight pipe, in inches, to be added for each 90-deg. elbow in the line

$$= \frac{76\ D_1}{\left(1 + \frac{3.6}{D_1}\right)}.$$

Where D_1 = inside diameter of pipe in inches. The values in Table 6 have been computed from the above formulas.

Example: Let it be required to determine the pressure loss in a pipe line for the following conditions:

$$D_1 = 5'' \qquad L = 300' \qquad w = 250 \text{ lb.}$$

Steam pressure = 150 lb. gage, or 165 lb. absolute.

$$d = \frac{1}{v} = \frac{1}{2.755} = 0.363 \text{ (from Table 5.)}$$

From Table 6, value of constant $A = 0.000,000,07$. By substitution in equation (8)

$$p = 0.000,000,07 \times \frac{250^2 \times 300}{0.363} = 3.62 \text{ lb. per sq. in.}$$

Steam Flow Chart. The use of *steam flow charts* based on the above formulas is very general in engineering practice, and a variety of these charts have been prepared using various coordinates depending on the relations which are to be expressed. Thus charts may be laid out to show velocity of flow, weight of steam, or pressure loss. The latter value is most often required in proportioning a piping system, and the following logarithmic chart, Fig. 18, by *Professor H. V. Carpenter* will be found very useful, as it shows the relation between size of pipe, average pressure, drop in pressure, and weight of steam passing in pounds per minute.

Examples. Follow the heavy dotted lines, and assume an allowable pressure loss of 0.3 lb. per 100 ft. for a 3-in. pipe at an average pressure of 80 lb. absolute. The weight of steam delivered will be 21 lb. per min. Again, assume a drop of 1 lb. per 100 ft. for a 10-in. pipe delivering 860 lb. per min. The average absolute pressure must be 60 lb. per sq. in. Finally, assume a 20-in. pipe is delivering 4,000 lb. per min. at an average absolute pressure of 250 lb. per sq. in. The drop in pressure will be 0.15 lb. per 100 ft. of pipe.

Professor Carpenter says, regarding the accuracy of the charts: "They represent the formulas exactly, except for the inaccuracies in drawing and in reading the scales. These errors are far within the limits of accuracy needed in practice so the charts may be used with the same degree of confidence as the formulas.

"As to the accuracy and range of the formulas, it seems that all the published experiments were made with pipes of from 1.85 to 4.0 in. in diameter. There is little doubt that the formulas may be applied with entire safety over a much wider range than this, but the practical limits are unknown."

Flow of Steam Through Orifices. The flow of steam from a higher to a lower pressure increases as the difference in pressure increases to a point where *the absolute terminal pressure becomes 58 per cent of the absolute initial pressure*. Below this point the flow is not increased by a reduction of the terminal pressure, even to the extent of a perfect vacuum. The lowest initial pressure for which this statement holds, when steam is discharged into the atmosphere, is 25.37 lb. For any pressure below this figure, the atmospheric pressure, 14.7 lb., is greater than 58 per cent of the initial pressure.

Napier deduced the following approximate formula for the flow of steam through an orifice.

$$W = \frac{p\,a}{70}.$$

Where W = the pounds of steam flowing per second,
p = the absolute pressure in pounds per square inch,
and a = area of the orifice in square inches.

In some experiments made by *Professor C. H. Peabody* on the flow of steam through pipes from ¼ in. to 1½ in. long and ¼ in. in diameter, with rounded entrances, the greatest difference from *Napier's* formula was 3.2 per cent excess of the experimental over the calculated results.

For steam flowing through an orifice from a higher to a lower pressure where the lower pressure is greater than 58 per cent of the higher, the flow per minute may be calculated from the formula:

$$W = 1.9\,A\,K\sqrt{(P - d)\,d}$$

Where W = the weight of steam discharged in pounds per minute,
A = area of orifice in square inches,
P = the absolute initial pressure in pounds per square inch,

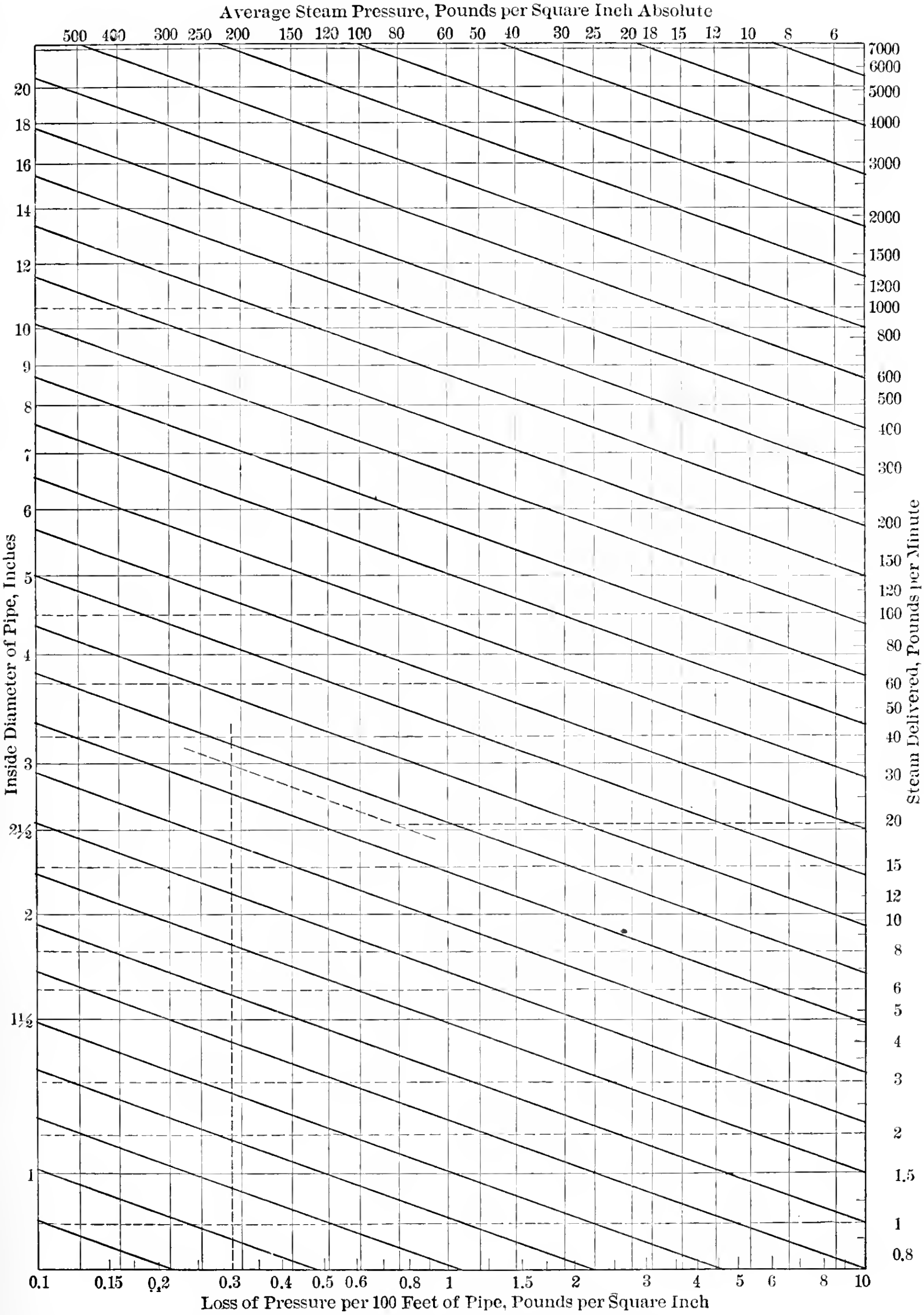


FIG. 18. CHART SHOWING LOSS OF PRESSURE WHEN A GIVEN AMOUNT OF STEAM PER MINUTE IS DELIVERED THROUGH A PIPE OF GIVEN SIZE.—*H. V. Carpenter.*

d = the difference in pressure between the two sides in pounds per square inch,
K = a constant = 0.93 for a short pipe, and 0.63 for a hole in a thin plate or a safety valve.

Example. Let it be required to determine the weight of steam flowing per min. from a boiler into the atmosphere through a short length of 1-in. pipe, for the following conditions:

Initial pressure in boiler (p) = 100 lb. absolute.

Internal area of 1-in. standard pipe (a) = 0.864 sq. in.

By substitution in *Napier's* formula

$$W = \frac{100 \times 0.864}{70} = 1.208 \text{ lb. per sec. or weight per min.} = 60 \times 1.208 = 72.48 \text{ lb.}$$

Measurement of Steam Flow. All steam meters for either indicating or recording the weight of steam flowing in a pipe are based on the following law:

$$W = A\, d\, V$$

in which

W = weight of steam flowing per sec.
A = internal area of pipe, sq. ft.
d = density of steam.
V = velocity, ft. per sec.

The density of steam is a function of the pressure and the quality, x, if it is wet saturated which is the usual condition in practice. The quality may be determined by means of a throttling or separating calorimeter previously described. The velocity in the Pitot tube type of meters, of which the *General Electric Co.'s* and the *Gebhardt* types are examples, is determined

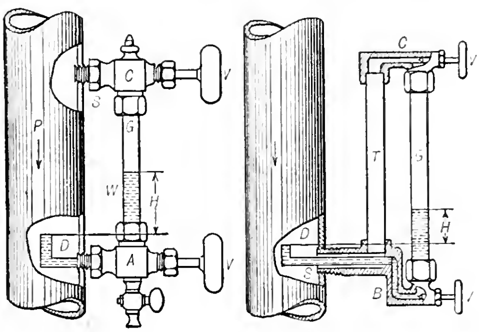


FIG. 19. PRINCIPLE OF THE PITOT-TUBE TYPE OF STEAM METER.

from the velocity head or pressure, measured by the height of a column of water or mercury supported by this head or pressure (Fig. 19).

The static head or pressure on the liquid column W is transmitted through the upper connection s while the total or dynamic pressure is transmitted to the liquid column by means of the tube D bent at right angles to the flow.

The height H of the liquid column is a measure of the difference between the total and static pressure, and is therefore an indication of the velocity head or pressure existing at the point of measurement. The relation between the height of the liquid in the tube and the velocity at the center of the pipe is determined from the following equation:

$$V = C\sqrt{2gh} \qquad (1)$$

in which

V = mean velocity of flow over entire cross section, ft. per sec.
h = height of a column in feet of the medium flowing.

C = a coefficient to correct for the average rate of flow as determined by experiment for various sizes of pipes.

The actual measurement of the velocity head is made in inches of water or mercury.

Let k = density of the liquid used in the tube.

d = density of the steam.

H = velocity head measured in inches of the liquid used in the tube or manometer.

$$12\,d\,h = k\,H \text{ or } h = \frac{kH}{12\,d}.$$

Substituting the value of h in (1)

$$V = C\sqrt{\frac{g\,k\,H}{6\,d}} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

The commercial form of this type of meter gives results within 2 per cent of actual condenser weights for velocity pressures corresponding to 1 inch or more of water.

The calibration of the indicating column to read the weight of steam flow direct is best made by weighing the water from a condenser to which the steam is delivered.

For a description of various forms of steam flow meters see *Carpenter and Diedrichs* "Experimental Engineering," also "Steam Power Plant Engineering" by *G. F. Gebhardt*.

AIR AND OTHER GASES

Properties of Air and Other Gases. Air is the most general example of a so-called perfect or permanent gas to be found in nature, and like the other so-called perfect gases conforms more or less closely to the laws of perfect gases. These laws are stated in the following paragraphs.

Pure dry air is a mechanical mixture of oxygen and nitrogen, that is, the oxygen and nitrogen can be separated from each other by purely physical means. This mixture is made up as follows:

	By Volume	By Weight
Oxygen........................	20.91%	23.15%
Nitrogen	79.09	76.85

Air as found in nature always contains other constituents in varying amounts such as carbon dioxide, ozone, water vapor, dust, bacteria, etc. See the chapter on "Ventilation and Air Analysis."

The *specific density*, or weight per cu. ft. of dry air decreases with the temperature, and conversely the *specific volume*, or volume per pound, which is always the reciprocal of the density, increases with the temperature. See Table 7 for properties of dry air.

The specific heat of air at constant pressure, or the B.t.u. required to raise one pound 1° F. at the pressure of the atmosphere, varies from 0.2375 to 0.2430 as determined by various investigators. The value 0.24 is recommended for engineering calculations.

It has been found that a given volume of air expands when heated under constant pressure, and again that if the temperature of a given volume of air is kept constant and the pressure increased, contraction takes place. These changes follow perfectly definite laws, which apply to other gases as well as air, known as "The Laws of Perfect Gases." These laws do not apply to steam, since it is not a perfect gas.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: With temperature constant, the volume of a given weight of gas varies inversely as its absolute pressure. Hence if P_1 and P_2 represent the initial and final absolute pressures and V_1 and V_2 represent corresponding volumes of the same mass, say 1 lb. of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$ or $P_1V_1 = P_2V_2$, but since P_1V_1 for any given case is a definite constant quantity, it

follows that the product of the absolute pressure and volume of a gas is a constant, or $PV = C$, when T is kept constant.

Any change in the pressure and volume of a gas at constant temperature, as indicated above, is called an *isothermal* change.

Charles' Law refers to the relation between pressure, volume, and temperature of a gas and may be stated as follows: The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume. Hence, when heat is added at constant volume V_c, we have the equation: $\frac{P_2}{P_1} = \frac{T_2}{T_1}$ or for the same temperature range, at constant pressure P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general we have for any weight of gas M, since volume is proportional to weight at any given volume and temperature, the relation

$$PV = MRT$$

which is the characteristic equation for a perfect gas. In this formula

P = the absolute pressure of the gas in pounds per square foot.
V = the volume of the weight M in cubic feet.
M = the weight in pounds of the gas taken.
R = a constant depending on the nature of the gas.
T = the absolute temperature in degrees F.

A perfect gas conforms exactly to the above equation, and while no gases are "perfect" in this sense they conform so nearly that the above equation will apply to most engineering computations.

Another form of the characteristic equation is sometimes used, in which M and R are eliminated. Let P_0, V_0, and T_0 denote the initial condition of a given quantity of a gas which undergoes a change in pressure, volume and temperature, the second condition being denoted by P, V, and T. For the initial condition then $P_0 V_0 = MRT_0$ or $\frac{P_0 V_0}{T_0} = MR$, and for the second condition $PV = MRT$ or $\frac{PV}{T} = MR$ so that the left hand members of the two equations are equal to each other, or $\frac{P_0 V_0}{T_0} = \frac{PV}{T}$.

So long as the same units are used for pressure, as pounds or ounces, and the same units are used for volume, as cu. ft. or cu. meters, and the temperatures are expressed in the same absolute scale it makes no difference what these units may be and the above equation holds.

In order to determine the value of R for any gas we must know the absolute pressure and temperature, and the volume in cu. ft. of one pound. For air at sea level, the absolute pressure is 14.7 lb. per sq. in or 2146.3 lb. per sq. ft. and at a temperature of 32° F. the absolute temperature is $32 + 459.6 = 491.6$ °F., and the volume is 12.39 cu. ft. per 1 lb. Now since $R = \frac{PV}{T}$ we have $R = \frac{2146.3 \times 12.39}{492} = 53.37$ *a constant for air.*

It follows then that the volume of 1 lb. of air (known as the specific volume) at any temperature and pressure, can be found at once by the equation $V = \frac{53.37 \times T}{P}$, and the value of R for other gases will be directly proportional to the specific volumes of such gases and air. See Table 8.

TABLE 7

PROPERTIES OF DRY AIR

Barometric Pressure 29.921 Inches

Temperature, Degrees Fahr.	Weight per Cubic Foot, Pounds	Per Cent of Volume at 70° F.	B.t.u. Absorbed by One Cubic Foot Dry Air per Degree F.	Cubic Foot Dry Air Warmed One Degree per B.t.u.
0	0.08636	0.8680	0.02080	48.08
5	.08544	.8772	.02060	48.55
10	.08453	.8867	.02039	49.05
15	.08363	.8962	.02018	49.56
20	.08276	.9057	.01998	50.05
25	.08190	.9152	.01977	50.58
30	.08107	.9246	.01957	51.10
35	.08025	.9340	.01938	51.60
40	.07945	.9434	.01919	52.11
45	.07866	.9530	.01900	52.64
50	.07788	.9624	.01881	53.17
55	.07713	.9718	.01863	53.68
60	.07640	.9811	.01846	54.18
65	.07567	.9905	.01829	54.68
70	.07495	1.0000	.01812	55.19
75	.07424	1.0095	.01795	55.72
80	.07356	1.0190	.01779	56.21
85	.07289	1.0283	.01763	56.72
90	.07222	1.0380	.01747	57.25
95	.07157	1.0472	.01732	57.74
100	.07093	1.0570	.01716	58.28
105	.07030	1.0660	.01702	58.76
110	.06968	1.0756	.01687	59.28
115	.06908	1.0850	.01673	59.78
120	.06848	1.0945	.01659	60.28
125	.06790	1.1040	.01645	60.79
130	.06732	1.1133	.01631	61.32
135	.06675	1.1230	.01618	61.81
140	.06620	1.1320	.01605	62.31
145	.06565	1.1417	.01592	62.82
150	.06510	1.1512	.01578	63.37
160	.06406	1.1700	.01554	64.35
170	.06304	1.1890	.01530	65.36
180	.06205	1.2080	.01506	66.40
190	.06110	1.2270	.01484	67.40
200	.06018	1.2455	.01462	68.41
220	.05840	1.2833	.01419	70.48
240	.05673	1.3212	.01380	72.46
260	.05516	1.3590	.01343	74.46
280	.05367	1.3967	.01308	76.46
300	.05225	1.4345	.01274	78.50
350	.04903	1.5288	.01197	83.55
400	.04618	1.6230	.01130	88.50
450	.04364	1.7177	.01070	93.46
500	.04138	1.8113	.01018	98.24
550	.03932	1.9060	.00967	103.42
600	.03746	2.0010	.00923	108.35
700	.03423	2.1900	.00847	118.07
800	.03151	2.3785	.00782	127.88
900	.02920	2.5670	.00728	137.37
1000	.02720	2.7560	.00680	147.07
1200	.02392	3.1655	.00603	165.83

Specific Heat of Gases. Reference has already been made to the fact that gases have two specific heats, one is the *specific heat at constant pressure* C_p and the other the *specific heat at constant volume* C_v.

The value of C_v can be found experimentally if we take one pound of gas occupying a fixed volume V_1 at pressure P_1. The absolute temperature is then $T_1 = \frac{P_1 V_1}{R}$. Now add heat to this gas and its temperature and pressure will become P_2 and T_2. No external work has been done as the volume remained constant and hence all the heat supplied has been used to raise the temperature of the gas. See Fig. 20. If H represents the heat added then $H = C_v (T_2 - T_1)$ or

$C_v = \dfrac{H}{(T_2 - T_1)}$, where C_v = specific heat at constant volume = B.t.u. required to raise 1 lb. of the gas 1° F.

TABLE 8

THERMAL PROPERTIES OF GASES

Name of Gas	Chem. Sym-bol	Mol. Weight O_2 = 32	Density Lb. per Cu. Ft. 32° F. & 1 Atmos.	Gas Constant R	$n = \frac{C_p}{C_v}$	Specific Heat	
						C_p	C_v
1	2	3	4	5	6	7	8
Air	.....	28.95	0.0807	53.34	1.40	0.240	0.171
Acetylene	C_2H_2	26.02	.0725	59.34	1.28	.350	.270
Ammonia* Superheated	NH_3	17.06	.0476	90.50	1.31	.523	.399
Argon	A	39.9	.1112	38.70	1.66	.124	.075
Carbon Dioxide*	CO_2	44.0	.1227	35.09	1.31	.210	.160
Carbon Monoxide	CO	28.0	.0780	55.14	1.41	.243	.172
Ethylene*	C_2H_4	28.02	.0780	55.08	1.20	.400	.330
Helium	He	4.0	.0112	386.0	1.66	1.250	.750
Hydrogen	H_2	2.016	.00562	765.86	1.40	3.420	2.440
Methane	CH_4	16.03	.0447	96.31	1.32	.593	.450
Nitric Oxide	NO	30.04	.0838	51.40	1.40	.231	.165
Nitrogen	N_2	28.08	.0783	54.99	1.40	.247	.176
Oxygen	O_2	32.0	.0892	48.25	1.40	.217	.155
Steam*	H_2O	18.016	....	85.72	1.28	.461	.351
Sulphur Dioxide*	SO_2	64.06	.1786	24.10	1.25	.154	.123

* Properties of these gases vary greatly with the temperature and pressure.

The value of C_p can also be found in a somewhat similar manner if we assume we have 1 lb. of gas in a cylinder fitted with a frictionless piston which is 1 sq. ft. in area. Initial condition is P_1, V_1, and T_1, where P_1 is the constant weight of the piston. Now add heat and change the

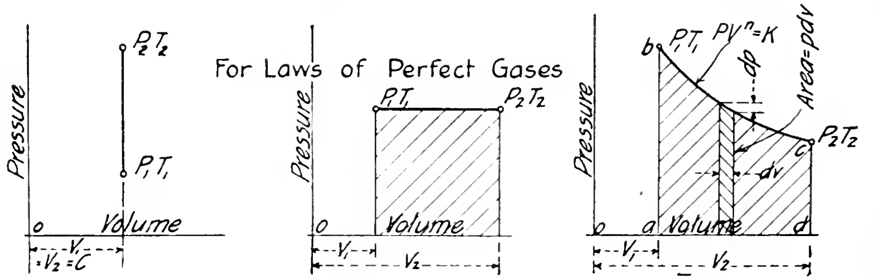


FIG. 20. FIG. 21. FIG. 22.

volume and temperature to V_2 and T_2, but $P_2 = P_1$. In this case we have performed external work by raising the piston, as well as increased the temperature of the gas. The work done is equal to $P_1 (V_2 - V_1)$ and its heat equivalent is found by dividing by 778. See Fig. 21.

If H represents the heat added then $H = C_v (T_2 - T_1) + \dfrac{P_1 (V_2 - V_1)}{778}$ = heat to change temperature + work done.

But if C_p = specific heat at constant pressure then $C_p (T_2 - T_1) = C_v (T_2 - T_1) + \dfrac{P_2 V_2 - P_1 V_1}{778}$ since $P_1 = P_2$. Also $P_2 V_2 = R T_2$ and $P_1 V_1 = R T_1$ so that $C_p (T_2 - T_1) =$

$C_v (T_2 - T_1) + \frac{R (T_2 - T_1)}{778}$ or $C_p = C_v + \frac{R}{778}$ for the gas in question. From this relation it will be seen that the specific heat at constant pressure is always greater than that at constant volume. See Table 8 for values of specific heats.

Expansion and Compression of Perfect Gases. The heat required to change the volume of a gas, the relation between the pressure and volume being expressed by some law, such as $P V^n = K$ (a constant) is found in the following manner. Referring to Fig. 22, it is apparent $P_1 V_1^n = K$ and $P_2 V_2^n = K$ from which

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{V_2}{V_1}\right)^n \text{ or } n = \frac{log\ P_1 - log\ P_2}{log\ V_2 - log\ V_1},$$ so that K can be readily found.

Now the total heat required will be that necessary to change the temperature $C_v (T_2 - T_1)$ and do the external work W represented by the area $a\ b\ c\ d$.

This area $abcd$ is equal to the summation of the elementary areas $P\,d\,V = \int_{v_1}^{v_2} P\,d\,V$, but $P = \frac{K}{V^n}$ so that $W = \int_{v_1}^{v_2} \frac{K\,d\,V}{V^n} = \frac{K}{n-1}\left[\frac{-1}{V^{(n-1)}}\right]_{v_1}^{v_2} = \frac{K}{n-1}\left[\frac{1}{V_1^{n-1}} - \frac{1}{V_2^{n-1}}\right]$. Now substitute the value of $K = P_1 V^n_1 = P_2 V^n_2$ in the last expression and we have $W = \frac{1}{n-1}$ $[P_1 V_1 - P_2 V_2]$ ft.-lb. or $W = \frac{1}{n-1}[R\,T_1 - R\,T_2] = \frac{R}{1-n}[T_2 - T_1]$ and expressed in heat units $= \frac{R}{778\,(1-n)}[T_2 - T_1]$. Hence the heat required is $H = \left[C_v + \frac{R}{778\,(1-n)}\right](T_2 - T_1)$.

Value of the exponent n. If expansion or contraction takes place without loss or gain of heat the change is said to be *adiabatic.* In this case no heat is added and hence $H = 0 = \left(C_v + \frac{R}{778\,(1-n)}\right)(T_2 - T_1)$. But as already stated $(C_p - C_v) = \frac{R}{778}$, and by substitution $C_v + \frac{C_p - C_v}{1-n} = 0$, or $C_v - n\,C_v + C_p - C_v = 0$. From which $n = \frac{C_p}{C_v}$ and hence the value of the exponent for adiabatic compression or expansion of a gas is equal to the ratio of the specific heats.

If we compress a gas adiabatically *the work of compression expressed in heat units is equal to the heat required to change the temperature.* As already stated $W = \frac{1}{n-1}(P_1 V_1 - P_2 V_2)$, the work of compression in ft.-lb. But for an adiabatic change $H = 0 = C_v (T_2 - T_1) + W \frac{1}{778}$ from which it appears that $\frac{W}{778} = C_v (T_1 - T_2)$.

Furthermore when a gas is expanded adiabatically the work performed by the gas expressed in heat units is equal to the heat abstracted in lowering its temperature.

The relation between pressure, volume and temperature, for adiabatic compression or expansion, can be expressed as follows, the value of n being $\frac{C_p}{C_v}$, and the initial and final states being P_1, V_1, T_1, and P_2, V_2, T_2. The characteristic equation of a perfect gas where M is 1 lb. can be stated as $T = \frac{P\,V}{R}$, and hence $\frac{T_1}{T_2} = \frac{P_1 V_1}{P_2 V_2} = \left(\frac{P_1}{P_2}\right) \times \left(\frac{V_1}{V_2}\right)$. Also, we have since

$P_1 V_1^n = P_2 V_2^n = K$ that $\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n$ and therefore $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{n-1}$ and $\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$ and $\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{\frac{n-1}{n}}$.

Those last three equations may be readily solved by the use of a table of logarithms.

Measurement of Air Flow. There are several methods employed for measuring the quantity of air delivered by a fan, blower or air compressor. The two methods most commonly employed in this connection are (1) by means of a circular orifice and (2) by the Pitot tube. The method employing the Pitot tube is fully described under the chapter on "Hot Blast Heating."

The Orifice Method. The discharge from the compressor or fan is piped to a gauging box similar in construction to the one shown in Fig. 23. The opposite end of the box is provided

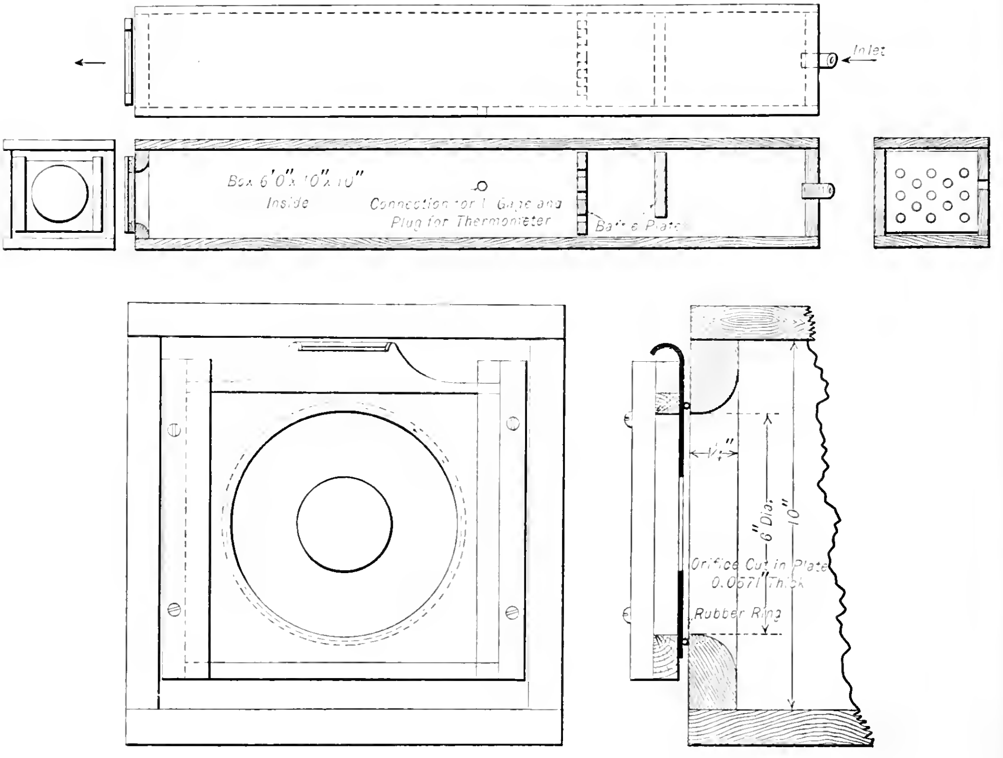


FIG. 23. DETAILS OF GAUGING BOX AND ORIFICE.

with a circular orifice as shown, discharging directly into the air. The static pressure existing within the box is measured by means of a U tube, which indicates the difference in pressure in inches of water between the two sides of the orifice. The temperature of the air passing through the gauging box is also recorded as well as the barometric pressure of the air. The discharge from the orifice must be free and unobstructed, so that the pressure on the discharge side will always be that of the atmosphere.

The *weight of air* passing the orifice per second is then readily determined by substituting in the following equation. The coefficient C to be used in this equation has been determined

by *R. J. Durley*, and may be taken from the curves in Fig. 24 for various sizes of orifices and differences in head. A complete discussion of this method of measuring air will be found in

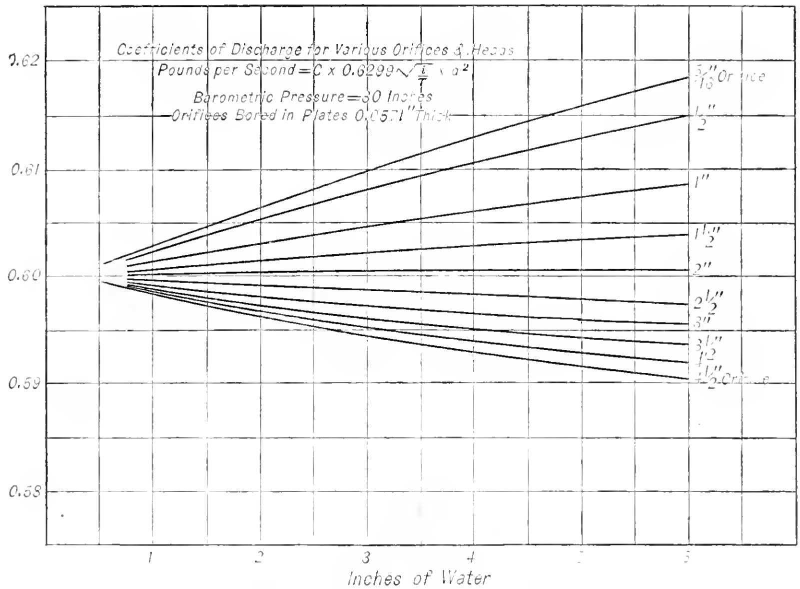


FIG. 24. COEFFICIENTS OF DISCHARGE FOR VARIOUS ORIFICES AND HEADS.

Vol. 27 of the "Transactions of the A. S. M. E." under "Air Flowing into Atmosphere through Circular Orifices."

$$W = 0.01369 \times C + d^2 \sqrt{\frac{i\,P}{T}}, \text{ in which}$$

W = weight of air flowing in pounds per sec.
C = a coefficient depending on values of d and i (see chart, Fig. 24).
d = diameter of orifice in inches.
i = difference in pressure in inches of water between the two sides of the orifice.
T = absolute temperature of the air passing the orifice = $460° + t°$.
t = degrees Fahrenheit in gauging box.
P = pressure in pounds per sq. ft. of the atmosphere based on barometer reading.

The above formula may be used for any atmospheric pressure, but for 30″ barometric pressure the formula reduces to:

$$W = 0.6299\ C\ d^2 \sqrt{\frac{i}{T}}.$$

Flow of Air through Pipes and Ducts. The same general formula as used for the flow of water may be applied, with sufficient accuracy, for air flowing under *low pressures* as in ventilating ducts, flues and chimneys.

The flow of air under low pressures is fully discussed under the chapter on "Hot Blast Heating."

TABLE 9

LOSS OF PRESSURE CAUSED BY FRICTION OF COMPRESSED AIR IN PIPES

Equivalent Volume in Cubic Feet Free Air per Minute Passing Through Pipe	Size of Pipe																		
	1″	1¼″	1½″	2″	2½″	3″	3½″		5″	6″	7″	8″	9″	10″	12″	14″	16″	18″	20″
	$P_1^2 - P_2^2$ = Difference of squares of initial and final absolute pressure, per 100 feet of pipe																		
50	125	40.9	16.5	3.9															
75	281.2	92.2	37.0	8.8															
100	500.0	164.0	65.8	15.6	5.1														
150	1125	368.6	148.1	35.2	11.5	4.6													
200	2000	655.4	263.4	62.5	20.5	8.2	3.8	1.9											
250	3125	1024	411.5	97.7	32.0	12.9	5.9	3.1											
300	4500	1475	592.6	140.5	46.1	18.5	8.6	4.4											
400	8000	2621	1053	250	81.9	32.9	15.2	7.8	2.6										
500	12500	4096	1646	390.6	128	51.4	23.8	12.2	4.0										
600	18000	5898	2370	562.5	184.7	74.1	34.3	17.6	5.8	2.3									
800		10486	4214	1000	327.7	131.7	60.9	31.3	10.2	4.1									
1000		16380	6584	1562	512.0	205.8	95.2	48.8	16.0	6.4	3.0								
1500		36860	14815	3516	1152	463.0	214.2	109.9	36.0	14.5	6.7								
2000			26339	6250	2048	823.0	380.8	195.3	64.0	25.7	11.9	6.1							
3000			59260	14050	4608	1852	856.8	439.5	144.0	57.9	26.8	13.7	7.6						
4000				25000	8192	3292	1523	781.2	256.0	102.9	47.6	24.4	13.6	8.0					
5000				39060	12800	5144	2380	1221	400.0	160.7	74.4	38.2	21.2	12.5	5.0				
6000				56250	18470	7407	3427	1758	576.0	231.5	107.1	54.9	30.5	18.0	7.2				
8000					32770	13170	6093	3125	1024	411.5	190.4	97.7	54.2	32.0	12.9	5.9			
10000					51200	20580	9520	4883	1600	640.0	297.5	152.6	84.7	50.0	20.1	9.3	4.8		
15000						46300	21420	10990	3600	1447	669.4	343.3	190.5	112.5	45.2	20.4	10.7	6.0	
20000						82300	38080	19530	6400	2572	1190	610.4	338.7	200.0	80.4	37.2	19.1	10.6	6.25
25000							59500	30520	10000	4019	1859	953.7	529.2	312.5	125.6	58.1	29.8	16.5	9.8
30000								43950	14400	5987	2677	1373	762.1	450	180.8	83.7	42.9	23.8	14.1
35000								59810	19600	7877	3644	1869	1037	612.5	246.2	113.9	58.4	32.4	19.1
40000								78120	25600	10288	4760	2441	1355	800.0	321.5	148.7	76.3	42.3	25.0
50000									40000	16070	7437	3815	2117	1250	502.4	232.4	119.2	66.2	39.1
60000									57600	23150	10710	5493	3048	1800	723.4	334.7	172.1	95.3	56.3
80000									102400	41150	10940	9766	5419	3200	1286	595.0	305.2	169.4	100
100000									160000	64000	29750	15260	8468	5000	2009	929.6	476.8	264.4	156.3

TABLE 10

PRESSURES AND SQUARES OF PRESSURES

Gage Pressure	Absolute Pressure	Square of Absolute Pressure	Gage Pressure	Absolute Pressure	Square of Absolute Pressure	Gage Pressure	Absolute Pressure	Square of Absolute Pressure	Gage Pressure	Absolute Pressure	Square of Absolute Pressure
0	14.7	216	...	.....	.....	...	.....	.....	....	......	....
2	16.7	279	...	.....	.....	...	.....	.....	....	......	....
4	18.7	350	...	.....	.....	...	.....	.....	....	......	....
6	20.7	428	...	.....	.....	...	.....	.....	....	......	....
8	22.7	515	...	.....	.....	...	.....	.....	....	......	....
10	24.7	610	56	70.7	4998	105	119.7	14328	240	254.7	64855
12	26.7	713	58	72.7	5285	110	124.7	15550	250	264.7	70055
14	28.7	824	60	74.7	5580	115	129.7	16822	260	274.7	75450
16	30.7	942	62	76.7	5883	120	134.7	18144	270	284.7	81050
18	32.7	1069	64	78.7	6194	125	139.7	19516	280	294.7	86845
20	34.7	1204	66	80.7	6512	130	144.7	20938	290	304.7	92840
22	36.7	1347	68	82.7	6839	135	149.7	22410	300	314.7	99040
24	38.7	1498	70	84.7	7174	140	154.7	23932	325	339.7	115400
26	40.7	1656	72	86.7	7517	145	159.7	25504	350	364.7	132940
28	42.7	1823	74	88.7	7868	150	164.7	27125	375	389.7	151850
30	44.7	1998	76	90.7	8226	155	169.7	28790	400	414.7	191950
32	46.7	2180	78	92.7	8593	160	174.7	30500	425	439.7	193300
34	48.7	2372	80	94.7	8968	165	179.7	32290	450	464.7	215925
36	50.7	2570	82	96.7	9351	170	184.7	34100	475	489.7	239790
38	52.7	2777	84	98.7	9742	175	189.7	35980	500	514.7	264900
40	54.7	2992	86	100.7	10140	180	194.7	37905	550	564.7	318900
42	56.7	3215	88	102.7	10547	185	199.7	39875	600	614.7	378900
44	58.7	3446	90	104.7	10962	190	204.7	41900	650	684.7	441800
46	60.7	3684	92	106.7	11385	195	209.7	43970	700	714.7	510800
48	62.7	3931	94	108.7	11816	200	214.7	46090	750	764.7	584800
50	64.7	4186	96	110.7	12254	210	224.7	50490	800	814.7	663750
52	66.7	4449	98	112.7	12701	220	234.7	55060	900	914.7	836700
54	68.7	4720	100	114.7	13156	230	244.7	59860	1000	1014.7	1029650

Flow of Compressed Air in Pipes. The variation of density with variation of pressure due to the elasticity of air makes a determination of the friction losses accompanying its passage through pipes a more complicated matter than the calculation for water-friction losses. Water being of practically constant density under all ordinary pressures, its rate of flow through a pipe of uniform diameter will be uniform throughout the length of that pipe, in spite of the decreasing pressure accompanying its progress. The friction losses through a unit distance—say 100 feet—in any part of the pipe line will therefore be the same as the loss through an equal distance in any other part of the pipe; or, in other words, the losses are directly proportional to the length of straight pipe. Air, on the other hand, enters a pipe at a certain pressure and velocity; as it advances through the pipe a certain loss of pressure occurs in overcoming frictional resistance: this loss of pressure is, however, accompanied by an increase of volume, and a corresponding increase in velocity of flow. This variation in velocity of flow throughout the length of the line results in a variation in frictional resistance, and the loss of pressure in a unit distance is the same at no two points in the pipe.

Table 9 is based on the formula of *J. E. Johnson, Jr.*, published in the *American Machinist*, July 27, 1899:

$$P_1^2 - P_2^2 = \frac{0.0006\,V^2 L}{D^5};$$

in which P_1 = absolute initial air pressure, lb.

P_2 = absolute terminal pressure air, lb.

V = free air equivalent in cubic feet per minute of volume passing through pipe.

L = length of pipe, feet.

D = diameter of pipe, inches.

The "free air equivalent" referred to above is the volume measured at atmospheric pressure.

CHAPTER III

HEAT TRANSMISSION OF BUILDING CONSTRUCTION

CONDITIONS AFFECTING HEAT TRANSMISSION

The amount of heat required to maintain the inside temperature of a building above that of the outside depends upon the following items, the first two of which are commonly termed *heat losses*, and each should be separately estimated and provided for in the design of any heating system.

Heat Required and Supplied. (*a*) The heat required to offset the heat transmission of the walls, ceiling, or roof, and the floor. This loss of heat depends upon the type and materials of construction used and the temperature difference to be maintained between the inside and outside.

(*b*) The heat required to warm the air entering the building from the outside as may enter by infiltration or be purposely introduced for ventilation.

(*c*) The heat supplied by persons, lights, machinery, and motors, which may be deducted from the sum of items (*a*) and (*b*) to obtain the net amount of heat to be supplied by the heating apparatus.

It is customary in all calculations connected with the design of heating installations to base the estimate on the amount of heat to be supplied per hour by the apparatus. The total heat to be supplied per hour is H = (item a) + (item b) − (item c) B.t.u. The method in use for the calculation of the various items above mentioned will now be taken up and discussed in the order given.

Temperatures. The inside temperature to be maintained and the air required for ventilation for various classes of work are fully discussed in the chapter on "Ventilation," to which the reader is referred.

The outside temperature for which the heating installation should be designed is fixed by the lowest outside temperature that is liable to continue for several days during the heating season. See Table 3. It is the practice of *Mr. N. S. Thompson*, of the office of the *Supervising Architect of the Treasury Department*, in designing the heating apparatus for buildings located in southern cities where the lowest recorded temperature occurs infrequently, and then only for a day or two, to base the heat loss calculations on a temperature 10° in excess of the lowest recorded temperature for the previous ten years.

In northern cities where the temperature goes below 10° F. and remains at or near 0° F. for several days, the calculations are based on the lowest recorded temperature for the previous ten years.

TABLE 1

USUAL INSIDE TEMPERATURES SPECIFIED

Public Buildings	68°–72° F.
Factories	65°
Machine Shops	60°–65°
Foundries, Boiler Shops, etc	50°–60°
Residences	70°
Bath Rooms	85°
Schools	70°
Hospitals	72°–75°
Paint Shops	80°

The inside temperatures usually specified vary with the type of building and the use to which the building is put as shown by Table 1. In many cases the various rooms may also be maintained at different temperatures.

The following is quoted from *Mr. F. R. Still's* article on "Shop Heating":

"There is often a wide variation of opinion as to the best temperature to maintain in cold weather. As a general proposition, in foundries and other buildings where the work is active and somewhat vigorous, a temperature of 50 to 55 degrees Fahrenheit is sufficient. In machine and wood-working shops, it should be from 60 to 65 degrees Fahrenheit. In shoe, clothing, and other factories employing help where the work is more or less sedentary, the temperature should be 68 to 72 degrees Fahrenheit.

"The basis for determining the amount of heating surface is the most difficult point to settle. In northern climates, the outside temperature occasionally drops down to 10 degrees below zero, and in Minnesota, Dakota, and Montana it goes even lower than 30 degrees below zero. If a plant is designed for these extremely low temperatures, it is too large for the conditions prevailing for ninety-five per cent of the year. It therefore becomes a question of very fine judgment as to just what should be the basis upon which to figure. The following table will give a fair idea of what the resultant temperatures will be under the varying outside atmospheric conditions, the basis being zero:

TABLE 2

RESULTING INSIDE TEMPERATURES

Class of Building	Temp. Inside, Degrees Fahr.	Temp. Outside, 30° Below 0° Fahr.	Temp. Outside, 20° Below 0° Fahr.	Temp. Outside, 10° Below 0° Farh.	Temp. Outside, 0° Fahr.	Temp. Outside, 10° Above 0° Fahr.	Temp. Outside, 20° Above 0° Fahr.	Temp. Outside, 30° Above 0° Fahr.
Foundries	50	22	32	41	50	57	63	70
Machine Shops	60	34	43	52	60	67	73	78
Clothing Shops	70	45	55	63	70	76	81	86
Paint Shops	80	59	67	74	80	85	89	92

"From the table it will be noted that for climates where the temperature does not drop below twenty degrees, an inside temperature based on zero outside, gives very fair average results, but for climates where the temperature may be from 30 degrees to 10 degrees for several days at a time the basis is not low enough. For such cases, if the basis is zero, the resulting temperature should be figured about ten degrees higher than normal, or the temperatures in the table given for machine shops should be taken for foundries, and the temperatures for clothing shops should be taken for machine shops.

"On the other hand, if the coldest temperature ever reached is zero, and the protracted cold spells are usually at outside temperatures of from ten to twenty degrees above zero, the basis for determining the size of the plant in zero weather would be forty degrees at zero for foundries, and then taking the tabulated temperatures of foundries for machine shops and the latter for clothing shops."

Hourly Variations in Heating Requirements. In addition to the lowest and average temperatures during the heating season for any given locality it may also be advisable to ascertain

from the hourly temperature records for a number of years past the average hourly temperatures for each month of the heating season so that the probable heating requirements for various parts of the day may be readily estimated. *Mr. R. P. Bolton* has prepared such a chart (Fig. 1) for New York city, which not only shows this variation but makes it possible to readily estimate the number of boiler horsepower required per hour, provided the total capacity is placed at the top of the horsepower scale at the left of the diagram. A method of determining the boiler horsepower is given under " Short Rules for Estimating Heat Losses."

TABLE 3

OUTSIDE TEMPERATURES

Lowest and Average Temperatures in the *United States.* (All stated in Fahrenheit Degrees and Compiled from *United States Weather Bureau Records.*)

State	City	Lowest	*Ave.	State	City	Lowest	*Ave.
Ala.	Mobile	-1	57.7	Neb.	North Platte	-35	34.6
	Montgomery	5	56.1		Lincoln	-29	35.8
Ariz.	Flagstaff	-21	34.8	Nev.	Carson City	-22	....
	Phoenix	22	58.9		Winnemucca	-28	37.9
Ark.	Fort Smith	-15	49.5	N. H.	Concord	-35	33.1
	Little Rock	-12	52.0	N. J.	Atlantic City	-7	41.6
Cal.	San Diego	32	57.2	N. Y.	Saranac Lake	-38	34.1
	Independence	10	48.7		New York City	-6	40.1
Col.	Denver	-29	38.4	N. M.	Roswell	-14	48.9
	Grand Jct	-16	39.2		Santa Fe	-13	38.0
Conn.	Southington	-19	36.3	N. C.	Hatteras	8	53.3
D. C.	Washington	-15	42.9		Charlotte	-5	49.8
Fla.	Jupiter	24	69.8	N. D.	Devil's Lake	-51	18.9
	Jacksonville	10	60.9		Bismarck	-44	23.5
Ga.	Savannah	8	57.2	Ohio	Toledo	-16	36.8
	Atlanta	-8	51.4		Columbus	-20	39.8
Idaho	Boise	-28	39.6	Okla.	Oklahoma	-17	47.1
	Lewiston	-18	42.5	Ore.	Baker City	-20	34.1
Ill.	Chicago	-23	35.9		Portland	-2	45.4
	Springfield	-22	39.0	Pa.	Pittsburgh	-20	40.8
Ind.	Indianapolis	-25	40.4		Philadelphia	-6	41.8
	Evansville	-15	44.1	R. I.	Providence	-9	37.5
Ia.	Sioux City	-31	32.1		Rock Island	-4	39.7
	Keokuk	-26	37.6	S. C.	Charleston	7	56.9
Kan.	Dodge City	-26	....		Columbia	2	53.5
	Wichita	-22	42.9	S. D.	Huron	-43	25.9
Ky.	Louisville	-20	45.0		Yankton	-32	31.2
La.	New Orleans	7	60.5	Tenn.	Knoxville	-16	47.0
	Shreveport	-5	55.7		Memphis	-9	50.7
Me.	Eastport	-21	31.1	Tex.	Corpus Christi	11	62.7
	Portland	-17	33.5		Fort Worth	-8	49.5
Md.	Baltimore	-7	43.3	Utah	Salt Lake City	20	39.7
Mass.	Boston	-13	37.2	Vt.	Northfield	32	27.8
Mich.	Alpena	-27	29.1	Va.	Cape Henry	5	48.6
	Detroit	-24	35.3		Lynchburg	-5	45.2
Minn.	Duluth	-41	25.5	Wash.	Seattle	3	44.3
	Minneapolis	-33	28.4		Spokane	-30	37.0
Miss.	Meridian	-6	53.9	W. Va.	Parkersburg	-27	41.9
	Vicksburg	-1	56.0		Elkins	-21	38.8
Mo.	Springfield	-29	43.0	Wis.	La Crosse	-43	31.2
	Hannibal	-20	39.7		Milwaukee	-25	32.4
Mont.	Havre	-55	27.7	Wyo.	Cheyenne	-38	33.7
	Helena	-12	30.9		Lander	-36	29.0

* Average is taken from October 1 to May 1.

Heat Transmission through Building Walls. (*a*) The amount of heat which must be supplied the interior of a building artificially warmed, or must be extracted when the building is refrigerated, depends largely upon the type of construction employed. The transfer of heat through building construction has been experimentally investigated, by the French physicist *Peclet,* and many other later experimenters. The laws governing the transfer of heat which *Peclet* stated have been the basis of practically all treatises that have since been written on this subject. The transfer of heat always takes place, or heat is said to flow or pass, from a warmer to a colder body.

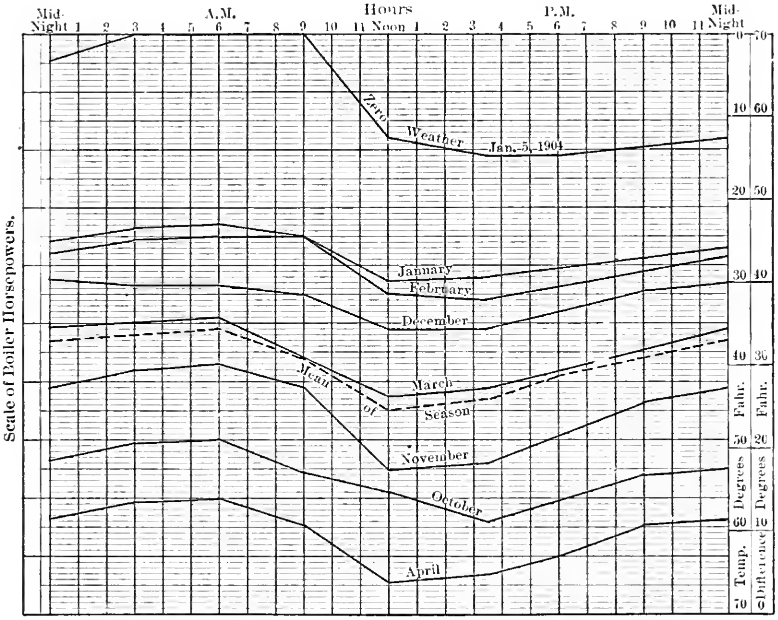


Fig. 1. Hourly Variation of Heating Requirements for New York City and Vicinity.

The division of the heating season in New York city is also given in the following table:

TABLE 4

DIVISION OF THE HEATING SEASON, NEW YORK CITY

Month	Days	Holidays	Sundays	Work-days
October	16	—	1.20	13.78
November	30	2	4.20	23.70
December	31	1	4.40	25.57
January	31	1	4.40	25.57
February	28.25	2	4.00	22.25
March	31	—	4.40	26.57
April	15	—	2.14	12.80
Total	182.25	6	24.74	150.24

Refer to Fig. 2 showing a section of a homogeneous wall, and let

t_0 = mean temperature of outside air.

t = mean temperature of inside air.

t_1 = temperature of inside wall surface.

t_2 = temperature of outside wall surface.

X = thickness of the wall in inches.

Heat will be transferred to the inside wall surface and be emitted by the outside wall surface in two ways. The so-called *radiant* heat passes in a straight line from the surface of the warmer body A, through the air, without appreciably heating it, to the receiving colder surface. The air in direct contact with the warmer body will absorb heat, and by the natural circulation, transfer and give it up in turn to the wall surface. This kind of heat transfer is termed *con-*

vection. The heat emission from the outer surface of the wall takes place in the reverse order. The transfer of heat from the inside to the outside wall surface through the material composing the wall is termed *conduction*.

Radiation. The quantity of heat which the surface of a material is capable of receiving or giving off to the surroundings by radiation is independent of the form, provided there are

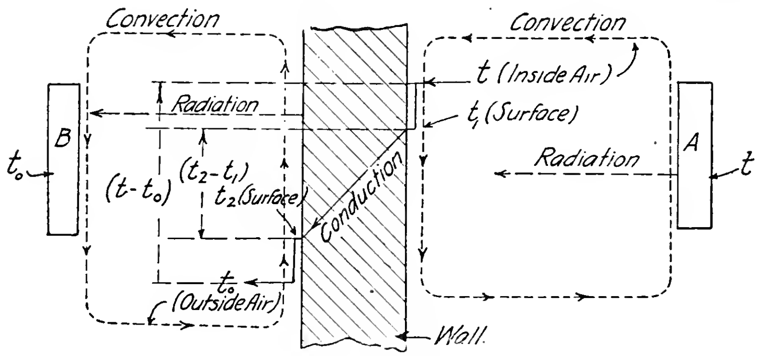


FIG. 2.

no re-entrant surfaces. It depends solely upon the nature of the surface and the absolute temperature of the surface in question and the absolute temperature of the object to or from which radiation is taking place.

The Stefan-Boltzman Radiation Law. The energy radiated from a black body is proportional to the difference of the fourth powers of the absolute temperatures of the radiating and receiving bodies, or

$$Q = D\left[\left(\frac{T_2}{100}\right)^4 - \left(\frac{T_0}{100}\right)^4\right] \text{ in which}$$

Q = B.t.u. radiated per sq. ft. per hour.
T_2 = Absolute temperature of the radiating body.
T_0 = Absolute temperature of the receiving body.
D = A constant.
= 0.1685 for a black body.

The radiation constant D for other than the black body depends upon the substance and the character of the radiating surface. The following values are taken from *Hütte:*

TABLE 5

RADIATION CONSTANTS

Material	Value of D
Glass, smooth	0.154
Brass, dull	.036
Copper, slightly polished	.0278
Wrought iron, dull	.154
Wrought iron, clean and bright	.056
Cast iron, rough	.157
Lime plaster, rough, white	.151
Slate	.115
Red sandstone finished smooth	.100

In applying the above formula and constants in practice it is necessary to make several assumptions which are of doubtful accuracy.

It is generally assumed that the temperature of the objects from which radiation takes place to the inside wall surface is the same as the inside air temperature, and that the temperature of the objects to which radiation takes place is the same as the outside air temperature.

The temperatures of the wall surfaces depend upon the temperature difference between the inside and outside air and the insulating value or conductivity of the material composing the wall.

At present there are no published data which give reliable information on the temperature differences referred to from which inside and outside wall temperatures may be determined.

Convection. The quantity of heat which may be transferred by convection or air contact, from a warmer to a colder surface, is independent of the form of the surface. It depends upon the difference in temperature between the surface and the mean temperature of the air in contact with it and also the rapidity of the circulation of the air over the surface.

A natural circulation of air exists within an artificially heated room due to the tendency of the warmer and less dense air to rise to the ceiling, while the air surrounding a building is often in rapid circulation. There are so many factors that affect the heat loss from the walls of a building by convection that it is quite impossible to give more than a rough approximation for this value.

Combined Coefficient of Radiation and Convection—K. Reliable experimental data are lacking for both the *radiation* and *convection coefficients* of the various materials of building construction. It is difficult to separate, in experimental work, the heat that is given off by radiation from that which is removed by convection. The combined heat loss due to both radiation and convection in practically still-air tests is, however, not difficult to obtain, and for the present, at least, furnishes the most satisfactory method of treating the problem.

The *combined coefficient* is defined as the heat absorbed or given off per square foot of surface per hour by radiation and convection under certain conditions of air movement, per degree difference in temperature between the surface and the average temperature of the air. If the air movement is different on the two sides of the wall, the value of the combined coefficient will of course be different owing to the fact that the heat loss by convection is different.

Let, K_1 = the combined coefficient for the inside wall surface.
K_2 = the combined coefficient for the outside wall surface.
$K_1\ (t - t_1)$ = the heat absorbed by the inside wall surface per sq. ft. per hour. (B.t.u.)
$K_2\ (t_2 - t_0)$ = the heat given off by the outside wall surface per sq. ft. per hour. (B.t.u.)
Then $K_1\ (t - t_1) = K_2\ (t_2 - t_0)$.
In which t and t_1 = temperature of the inside air and inside wall surface respectively.
t_0 and t_2 = temperature of the outside air and outside wall surface respectively.

The following average values of K_1 (Table 6) were determined from a series of tests made under the direction of the authors.* The tests were run under practically still air conditions, the only movement of the air being that due to the natural currents existing in the room in which the tests were conducted.

TABLE 6

VALUES OF K STILL AIR FROM AUTHORS' TESTS

Brickwork	1.35 to 1.40	Glass Window	1.50
Concrete	1.30	Sheet Asbestos	1.40
Corkboard	1.25	Magnesia Board	1.45
Cement Plaster Finish	0.93	Wood (finished surface)	1.40

* The tests referred to were conducted by *L. C. Lichty, Univ. of Ill.*, 1915.

The average value of K_1 from above data is 1.34. The value of K increases with the velocity of air over the surface. The value of K_2, for brickwork and wood, for various velocities of

air or wind movement may be obtained by multiplying the values of K_1 from the above table by the factors given in Table 7.

TABLE 7

MULTIPLIERS FOR DETERMINING K_2

Velocity, Miles per Hour	Multipliers of K_1	
	Brickwork	Wood
5	2.38	2.19
10	3.20	2.71
15	3.76	2.95
20	4.22	3.02

In practice the exposed walls of a building are not subjected to an average wind movement of more than 15 miles per hour in extremely cold weather. The authors, in their own practice, have adopted the general rule that the value of K_2 for an outside wall surface may be considered as being equal to three (3) times that of the inside wall surface.

The heat transmission of walls calculated in this manner gives results that are in accord with the general practice of heating and ventilating engineers.

Conductivity. The amount of heat that will be transmitted through a material having parallel surfaces, due to a difference in temperature between these surfaces, is termed the conductivity of the material. The amount of heat that a given material will transmit is directly proportional to the difference in temperature between the surface and inversely proportional to the thickness.

Let C = coefficient of conductivity, or B.t.u. transmitted per sq. ft. per hour per inch of thickness per degree Fahrenheit difference in temperature of the two surfaces.

t_1 = temperature of inside surface.

t_2 = temperature of outside surface.

X = thickness of wall inches.

Then $\frac{C}{X}(t_1 - t_2)$ = heat transmitted by conduction per sq. ft. per hour.

It is obviously impossible to give a table of exact conductivities of the various materials of building construction, owing to the fact that two samples of the same kind of material will often be found to vary considerably both in density and conductivity.

The following table gives the results of tests conducted under the direction of the authors.

TABLE 8

COEFFICIENT OF CONDUCTIVITY C FROM AUTHORS' TESTS

B.t.u. transmitted per sq. ft. per hour per inch thickness per deg. F. difference in temperature of the two surfaces. Materials thoroughly dry

	Wt. per Cu. Ft.	C
*Brickwork	132. lb.	4.00-(5.00)
Concrete (Stone 1. 2. 4 mix.)	140. "	8.30
Wood (Fir) $\frac{7}{8}$" thick	33.4 "	1.00
Corkboard Insulation	9.7 "	0.32
Corrugated Asbestos Board	20.4 "	0.48
Sheet Asbestos	48.3 "	0.29
Magnesia Board	13.5 "	0.51

* It is recommended that a value of C = 5. be used in the calculation for the heat transmission of brickwork to allow for an increased conductivity due to the possible presence of moisture.

The following figures are the conductivities for the thickness stated. (B.t.u. per sq. ft. per hour per deg. difference in temp. of the two surfaces.)

Glass (0.085″)	24.3
Window (76.3% glass)	8.64
Double Window with ½″ air space	1.04
2″ Hollow Tile plastered both sides	0.99
4″ Hollow Tile plastered both sides	0.61
6″ Hollow Tile plastered both sides	0.47
2″ Hollow Tile plastered both sides with ready prepared gravel roofing applied to one side only	0.84

The following additional values of C are quoted from various sources.

Concrete (Stone 1. 2. 4 mix.) (*Norton*)	6.25
Concrete (Cinder 1. 2. 4 mix.) (*Norton*)	2.35
Brickwork (*Poensgen*)	3.42
Sandstone (*Poensgen*)	9.00
Packed Granulated Cork (6.25 lb. per cu. ft.)	0.35
Packed Mineral Wool (16.3 lb. per cu. ft.)	0.35
Mortar	8.00

Calculation for Heat Transmission of Walls. The amount of heat received by the inside wall surface, the amount conducted through the wall, and the amount emitted by the outside surface must evidently be equal to one another.

Let u = the heat transmission of the actual wall per sq. ft. per hr. per deg. difference in temp. of the air on the two sides.

$$u\,(t - t_0) = K_1\,(t - t_1) = \frac{C}{X}\,(t_1 - t_2) = K_2\,(t_2 - t_0) \qquad (1)$$

$$\therefore u = \frac{1}{\dfrac{1}{K_1} + \dfrac{1}{K_2} + \dfrac{X}{C}} \qquad (2)$$

The value of $\dfrac{X}{C}$ for thin metal plates, building paper or glass is so small that it may safely be neglected in the calculations. If the wall is composed of several layers of different materials in contact with one another (no air spaces) then

$$u = \frac{1}{\dfrac{1}{K_1} + \dfrac{1}{K_2} + \left(\dfrac{X_1}{C_1} + \dfrac{X_2}{C_2} + \dfrac{X_3}{C_3} + \text{etc.}\right)} \qquad (3)$$

in which X_1, X_2, X_3, etc., are the thicknesses of the various materials in inches, C_1, C_2, C_3, etc., are the corresponding coefficients of conductivity, and K_1 and K_2 are the combined coefficients of radiation and convection for the inside and outside wall surfaces.

Values of u for a variety of building materials are given in Tables 9 and 10.

Example. The following examples will serve to illustrate the method employed in calculating the heat transmission of various materials, as given in Table 9:

13″ Brick Wall. Fig. 3.

$K_1 = 1.4 \quad K_2 = 3 \times K_1 = 4.2 \quad C = 5 \quad X = 13.$

$$u = \frac{1}{\dfrac{1}{1.4} + \dfrac{1}{4.2} + \dfrac{13}{5}} = 0.281.$$

Assuming an inside air temperature $t = 70°$ and an outside temperature $t_0 = 0$ the heat loss under these conditions will be $0.28 \times (70 - 0) = 19.7$ B.t.u. per sq. ft. of surface per hour.

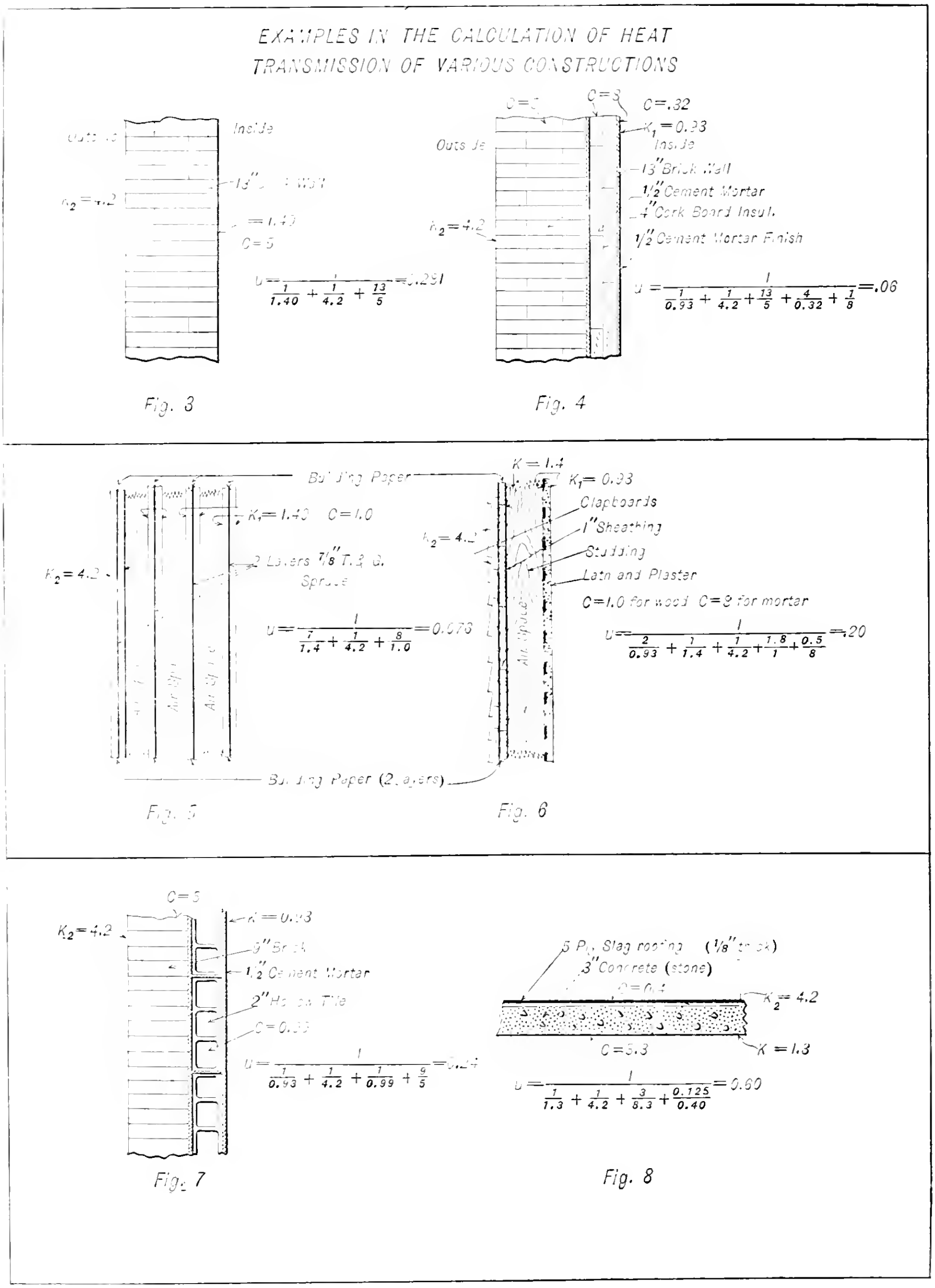
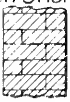
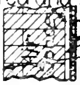
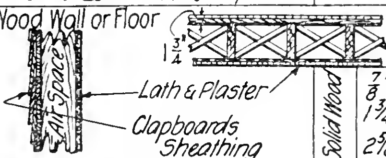
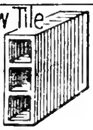
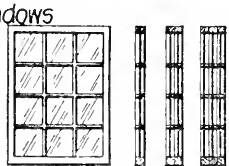

EXAMPLES IN THE CALCULATION OF HEAT
TRANSMISSION OF VARIOUS CONSTRUCTIONS
Inside
K_2 = 4.2
C = 5
U = 1 / (1/1.40 + 1/4.2 + 13/5) = 0.291
Fig. 3
C = .32
K_1 = 0.93
Inside
Outside
13" Brick Wall
1/2" Cement Mortar
4" Cork Board Insul.
1/2" Cement Mortar Finish
K_2 = 4.2
U = 1 / (1/0.93 + 1/4.2 + 13/5 + 4/0.32 + 1/8) = .06
Fig. 4
Building Paper
K_1 = 1.40 C = 1.0
K_2 = 4.2
2 Layers 7/8" T. & G. Spruce
U = 1 / (7/1.4 + 1/4.2 + 8/1.0) = 0.078
Air Space
Building Paper (2 Layers)
Fig. 5
K = 1.4
K_1 = 0.93
Clapboards
1" Sheathing
Studding
Lath and Plaster
K_2 = 4.2
C = 1.0 for wood C = 8 for mortar
U = 1 / (2/0.93 + 1/1.4 + 1/4.2 + 1.8/1 + 0.5/8) = .20
Air Space
Fig. 6
C = 5
K = 0.93
K_2 = 4.2
9" Brick
1/2" Cement Mortar
2" Hollow Tile
C = 0.99
U = 1 / (1/0.93 + 1/4.2 + 1/0.99 + 9/5) = 0.24
Fig. 7
5 Ply Slag roofing (1/8" thick)
3" Concrete (stone)
K_2 = 4.2
C = 8.3
K = 1.3
U = 1 / (1/1.3 + 1/4.2 + 3/8.3 + 0.125/0.40) = 0.60
Fig. 8

TABLE 9

HEAT TRANSMISSION OF BUILDING CONSTRUCTION

BASED ON TESTS BY THE AUTHORS

Construction	Thickness	B.t.u. transmitted per square foot per hour. Temperature difference 1°	20°	40°	60°	70°	80°
Plain Brick Wall.  $K_1 = 1.40$ $K_2 = 4.20$ $C = 5.0$	9"	.363	7.3	14.5	21.8	25.4	29.0
	13"	.281	5.6	11.2	16.9	19.7	22.5
	18"	.220	4.4	8.8	13.2	15.4	17.6
	24"	.174	3.5	7.0	10.4	12.2	13.9
Brick Wall + Air Space, Furred and Plastered  $K_1 = .93$ $K = 1.4$ $K_2 = 4.2$ $C_a = 5$ $C_{pl} = 8$	9"	.217	4.3	8.7	13.0	15.2	17.4
	13"	.185	3.7	7.4	11.1	13.0	14.8
	18"	.156	3.1	6.2	9.4	10.9	12.4
	24"	.132	2.6	5.3	7.9	9.2	10.6
Wood Wall or Floor  $1\frac{3}{4}$" Air Space, Lath & Plaster, Clapboards, Sheathing		.20	4.0	8.0	12.0	14.0	16.0
Solid Wood	$\frac{7}{8}$"	.547	10.9	21.9	32.8	38.3	43.8
	$1\frac{3}{4}$"	.370	7.4	14.8	22.2	25.9	29.6
	$2\frac{5}{8}$"	.279	5.6	11.2	16.7	19.5	22.3
Hollow Tile  $\frac{1}{2}$" Plaster on both sides $K_1 = .93$ $K_2 = 2.79$ $C = \{2"-.99;\ 4"-.61;\ 6"-.47\}$	2"	.409	8.2	16.4	24.5	28.6	32.7
	4"	.325	6.5	13.0	19.5	22.8	26.0
	6"	.281	5.6	11.2	16.9	19.7	22.5
Concrete  $K_1 = 1.30$ $K_2 = 3.90$ $C = 8.0$	2"	.784	15.7	31.4	47.0	54.9	62.7
	3"	.714	14.3	24.6	42.8	50.0	57.0
	4"	.655	13.1	26.2	39.3	45.9	52.4
	6"	.563	11.3	22.5	33.8	39.4	45.0
	For 3" concrete covered with slag roofing deduct approximately 10% from values stated						
Windows  $K_1 = 1.5$ $K_2 = 4.5$	Single	1.126	22.5	45.0	67.6	78.8	90.0
	Double	.450	9.0	18.0	27.0	31.5	36.0
	Triple	.281	5.6	11.2	16.9	19.7	22.5
Infiltration Loss - B.t.u. per hour = cu. ft. {leaking in at 70°} x {.075 x .24 x $(t-t_0)$}							
1 Air change per hour. Temperature of outside air 0° F, inside air 70° F. - Cu. ft x		.018	.360	.720	1.08	1.26	1.44
P = perimeter of window in feet. B.t.u. loss per hour per degree diff. = P x 60 x 13 x 0.147 x .086 x 0.24 = 2.4P for $\frac{1}{16}$" crack - Weather stripped sash = 0.15P	$\frac{1}{32}$" crack	1.2	24	48	72	84	96
	$\frac{1}{16}$" "	24	48	96	144	168	192
	Note: For this method use perimeters of windows on one side of room only						
Carpenter's Rule for calculating heat loss of buildings:- B.t.u. per hr. = $(G + \frac{1}{4}W + .02NC)(t - t_0)$ W = wall surface; C = cubic contents; G = glass surface, N = number of air changes per hour.							

NOTE.—The volume of air leaking in is measured at a temperature of 70°.

TABLE 10

B.T.U. TRANSMITTED PER HOUR PER SQUARE FOOT SURFACE

(*Buffalo Forge Co.*)

Kind of Surface	Thickness	1°	50°	60°	70°
Brick wall bricks $8\frac{1}{4}'' \times 4'' \times 2''$ with vertical mortar joints $\frac{3}{8}''$ thick	$8\frac{1}{2}''$	.37	18.5	22.2	25.9
	13″	.29	14.5	17.4	20.3
	$17\frac{1}{2}''$	.25	12.5	15.0	17.5
	22″	.22	11.0	13.2	15.4
	$26\frac{1}{2}''$	.19	8.5	11.4	13.3
Brick wall with $\frac{3}{4}''$ of plaster on one side	$8\frac{1}{2}''$	.36	18.0	21.6	25.2
	13″	.28	14.0	16.8	19.6
	$17\frac{1}{2}''$	.24	12.0	14.4	16.8
	22″	.21	10.5	12.6	14.7
	$26\frac{1}{2}''$	.18	9.0	10.8	12.6
Brick wall 4″ air space $\frac{3}{4}''$ plaster	8″	.25	12.5	15.0	17.5
	$12\frac{1}{2}''$	.21	10.5	12.6	14.7
	17″	.19	9.5	11.4	13.3
	$21\frac{1}{2}''$	.16	8.0	9.6	11.2
	26″	.14	7.0	8.4	9.8
Brick wall furred and $\frac{3}{4}''$ plaster	4″	.28	14.0	16.8	19.6
	$8\frac{1}{2}''$	.23	11.5	13.8	16.1
	13″	.20	10.0	12.0	14.0
	$17\frac{1}{2}''$	.18	9.0	10.8	12.6
	22″	.16	8.0	9.6	11.2
Frame walls: Clapboard 7/16 Studding Plaster		.44	22.0	26.4	30.8
Clapboard 7/16″ Paper Studding Plaster		.31	15.5	18.6	21.7
Clapboard 7/16″ Sheathing $\frac{3}{4}''$ Studding Plaster		.28	14.0	16.8	19.6

Kind of Surface		1°	50°	60°	70°
Clapboard 7/16″ Paper Sheathing $\frac{3}{4}''$ Studding Plaster		.23	11.5	13.8	16.1
Single Window...........		1.09	51.5	65.4	76.3
Double Window........		.46	23.0	27.6	32.2
Single Skylight..........		1.16	53.0	60.6	81.2
Double Skylight.........		.48	24.0	28.8	33.6
Doors or wooden walls (Pine)	1″	.41	20.5	24.6	28.7
	$1\frac{1}{2}''$	.32	16.0	19.2	22.4
	2″	.27	13.5	16.2	18.9

Kind of Surface	1°
Single Flooring: For Ceiling	.10
Single Flooring: For Floor	.07
Double Flooring: For Floor	.09
Double Flooring: For Ceiling	.06
Corrugated Iron and Concrete with Double Flooring	.13
Floor, single: no plaster beneath joists..........	.45
Floor, single $\frac{3}{4}''$: lath and plaster beneath joists..	.26
Floor, double $1\frac{1}{2}''$: no plaster beneath joists.....	.31
Floor, double $1\frac{1}{2}''$: lath and plaster beneath joists.	.18
Ordinary stud partition: lath and plaster one side....................................	.60
Ordinary stud partition: lath and plaster both sides....................................	.34

NOTE.—Thicknesses of brick walls refer to actual brick, exclusive of air spaces, plaster, furring, etc.

Single Glass Window.

$$K_1 = 1.5 \quad K_2 = 3 \times 1.5 = 4.5 \left(\frac{x}{c} \text{ may be neglected}\right)$$

$$u = \frac{1}{\frac{1}{1.5} + \frac{1}{4.5}} = 1.125.$$ For a 70° difference in temperature between the inside and outside the heat loss will be: $1.125 \times 70 = 78.8$ B.t.u. per sq. ft. per hour.

Other Types of Wall. The calculations for several other types of construction are shown by Figs. 3, 4, 5, 6, 7 and 8.

Heat Transmission of Air Space Construction. Heat is transmitted through an air space construction, from one surface to another by radiation and convection.

The calculations for wall constructions which contain air spaces, as, for example, the wood wall construction shown by Fig. 6, may be made as follows:

K_1 for the inside plastered wall surface $= 0.93$.
K for the inside surface of sheathing $= 1.40$.
K_2 for the outside wood clapboards $= 3 \times 1.4 = 4.2$.
$C = 1$ for wood, $C = 8$ for plaster.
The total thickness of the wood is approximately 1.8″ (average). Thickness of plaster ½″.

$$u = \frac{1}{\frac{2}{0.93} + \frac{1}{1.4} + \frac{1}{4.2} + \frac{1.8}{1} + \frac{0.5}{8}} = 0.20.$$

Table 10 of heat transmission coefficients and factors as given by the *Buffalo Forge Co.* represents a fair average of the values used by engineers in commercial practice to-day.

Determination of the Heat Transmission of Building Construction by Experiment. Of the several laboratory methods that are used to determine the heat transmission of building construction it is the opinion of the authors that the following is the most satisfactory and accurate:

A box is constructed (Fig. 9) of the material to be tested, inside of which is placed a resistance coil or bank of incandescent lamps, used as a heater, and a small disc fan to provide a circulation of the air in order to maintain a uniform inside temperature. The inside and outside air temperatures are measured in the usual manner. The temperature of the wall surfaces are most accurately determined by means of a thermocouple. The total heat introduced into the box is the sum of the heat equivalent of the watts supplied the resistance coil or lamps and the fan.

Let A_1 = current (amperes) supplied coil.
A_2 = current (amperes) supplied fan.
V_1 = voltage across terminals of coil.
V_2 = voltage across terminals of fan.
$W_1 = A_1V_1$ = watts supplied coil.
$W_2 = A_2V_2$ = watts supplied fan.
t = inside temperature of air in box, degs. F.
t_0 = outside temperature of air, degs. F.
u = heat transmission in B.t.u. per sq. ft., per hr., per deg. difference between inside and outside air temperatures.
S = mean heat transmitting surface of box in square feet.

$$1 \text{ watt hour} = \frac{33000 \times 60}{746 \times 778} = 3.415 \text{ B.t.u. per hour,}$$ as 746 watt hours = 1 horsepower hour.

$$\therefore u = \frac{3.415\,(W_1 + W_2)}{S\,(t - t_0)}.$$

NOTE.—The values of C and K for still air are readily determined if the temperatures of the surfaces are recorded.

It is not necessary to construct an entire box of the material to be tested each time. A standard box having once been thoroughly tested and the transmission factor u_1 accurately determined, one side is removed and the new material, for which the heat transmission factor is desired, substituted. The difference between the calculated amount of heat that would have

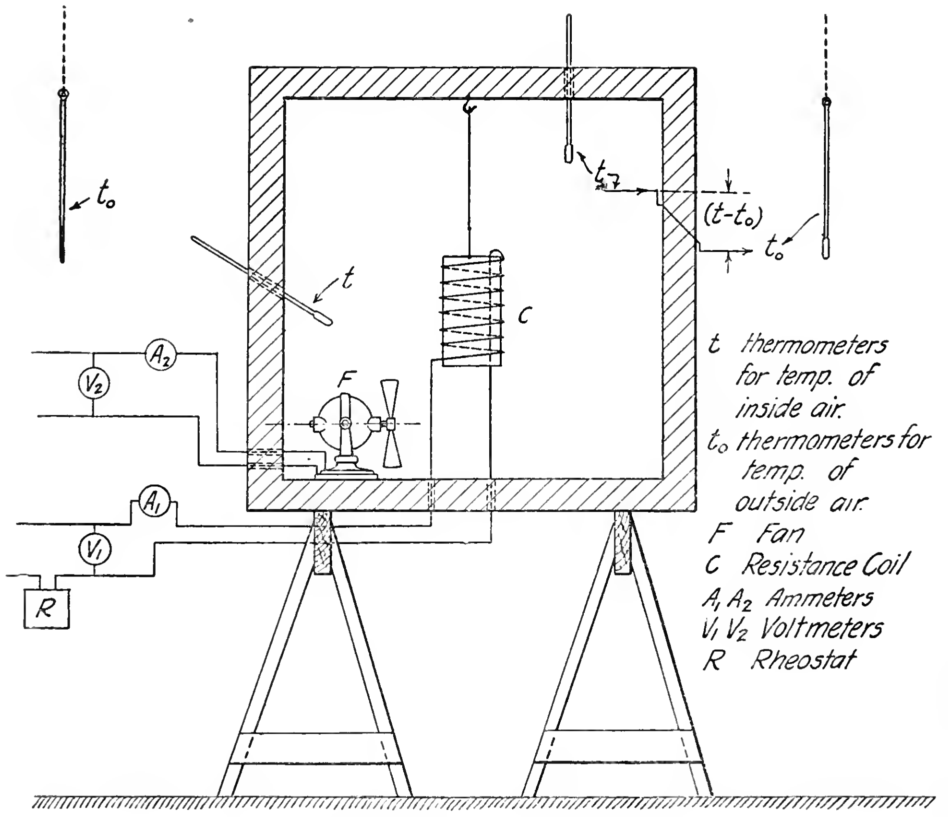


FIG. 9. TRANSMISSION TESTS OF BUILDING MATERIALS.

been required for five sides of the original test box and the actual heat input for the box with the substituted side gives the total heat transmission of the new material. This quantity divided by the product of the area of the substituted side and the temperature difference gives the heat transmission factor u_x of the material in question, as shown by the following calculation:

$$S = 6\,A \text{ and } W = W_1 + W_2 = \text{Watts to coil and fan.}$$

$$3.415\,W = (t - t_0)\,(5\,A\,u_1 + A\,u_x),$$

$$u_x = \frac{3.415\,W - 5\,A\,u_1\,(t - t_0)}{A\,(t - t_0)}$$

where A = area of one side in sq. ft.

Increase in the Heat Transmission Loss Due to High Ceilings. The specified room temperature (t) refers to the temperature of the air measured at the breathing line, or approximately five feet above the floor. This temperature may be used as the average temperature of the air

in contact with the walls for rooms that do not exceed 12 feet in height. When the height is greater than this the average temperature of the air in contact with the wall surfaces will be greater and a suitable correction should be made.

Rietschel states that the average mean temperature t_m never exceeds 1.15 t for the highest ceilings. His formula is $t_m = t\,[1. + 0.017\,(h - 10)\,]$, in which h is the height of the ceiling or outside walls. Substituting $t_m = 1.15\ t$, we find $h = 19$ ft., so that for heights over 19 ft. a value of $t_m = 1.15\ t$ may be used.

The following table of mean temperatures may be used in calculations based on the above formula:

TABLE 11

MEAN TEMPERATURES FOR ROOMS WITH HIGH CEILINGS

Height Feet, h	Average Temperature t_m		
	$t = 55°$	$t = 65°$	$t = 70°$
10′	55	65	70
15′	59.7	70.5	76
20′ and above	64.4	76.1	81.9

Exposure Factor. It is the practice of many engineers to increase the calculated heat transmission loss of the wall and glass surfaces by a certain percentage due to the fact that when the wall surfaces are exposed to high winds the convection factor is increased, and also to compensate for the cold air leakage through the wall itself.

The following more or less arbitrary factors, by which the calculated heat transmission loss of the walls is to be multiplied, are quite commonly used:

North, northeast, and northwest exposure where winds are to be counted upon as an important factor . . . . = 1.15 to 1.30
East or west walls moderately exposed . . . . = 1.10 to 1.20
South walls . . . . = 1.

It should be understood when the data from Table 9 are used it is not necessary to use an *exposure factor*.

Heat Transmission of Roofs and Floors. The temperature of the air in contact with the under side of a ceiling or roof is found to be higher than the temperature maintained at the breathing line, at which point the temperature is usually measured, due to the natural tendency of the warmer or less dense air to rise.

It is recommended that an increase of approximately 15 per cent be made to the specified inside temperature for the temperature at the ceiling for ceiling or wall heights not exceeding 15 ft., and 30 per cent for ceiling heights of 20 ft. or more in estimating the heat loss of roofs. Thus, if 65° is the specified inside temperature to be maintained in a room the height of which is 20 ft. the temperature of the air in contact with the under side of the roof may be assumed as 65° + 30 per cent or 85°. The loss of heat through the ceiling of a room over which a large air space exists, through partitions between a heated and cold room, or through the first floor to cellar, may be estimated on the assumption that the warmed rooms give off sufficient heat to maintain the temperature of these colder spaces according to the following schedule. These data are quoted from *Kinealy* as the practice of German engineers:

Closed attics under metal or slate roofs . . . . 14° F.
Closed attics under tile, cement, tar, or gravel roofs . . . . 23° F.
Cellars and rooms kept closed . . . . 32° F.

Thus the estimated heat loss from a room having a $\frac{7}{8}''$ wood floor located over an unheated cellar assuming a room temperature of 70° F. would be:

$$K = 1.4 \text{ both sides, } C = 1.0, \text{ and } u = 0.43$$
$$u\,(70 - 32) \text{ or } 0.43 \times 38 = 16.3 \text{ B.t.u. per sq. ft. per hour.}$$

The heat transmission of floors that are laid directly upon the ground may be estimated on the assumption that the ground in contact with the under side of the floor will have an approximate temperature of 50° F. Thus the estimated heat loss through a 6″ concrete floor laid directly upon the ground, assuming an inside temperature of 65° F., will be:

$$u = \frac{1}{\dfrac{1}{1.30} + \dfrac{6}{8}} = 0.66$$ B.t.u. per sq. ft. per hour per degree difference in temperature ($K = 1.30$ and $C = 8.0$). The estimated transmission loss per sq. ft. per hour is then

$$0.66 \times (65 - 50) = 9.9 \text{ B.t.u.}$$

Heat Loss by Infiltration. (*b*) The heat required to warm the outside air which may enter by *leakage* through the cracks or clearances around windows and doors or through the walls themselves, is that required to raise the temperature of the weight of incoming air per hour from the outside to the inside temperature.

Let b = B.t.u. required per hour to heat the incoming air.
t = inside room temperature degs. F.
t_0 = outside temperature.
C_p = sp. heat of air at constant pressure.
= 0.24.
d = density of the air, temperature t.
= 0.075 for 70°.
= 0.076 for 60°.
Q = cubic feet of air per hour entering building by infiltration, measured at temperature t.
W = weight of air per hour entering building by infiltration.
$= d \times Q$.
Then $b = C_p\,(t - t_0)\,Q \times d = 0.24 \times W \times (t - t_0)$.
$= 0.018 \times Q\,(t - t_0)$.
For $t_0 = 0$ and $t = 70°$ $\quad b = 1.26\,Q$.
For $t_0 = 0$ and $t = 60°$ $\quad b = 1.08\,Q$.

As the same weight of air in a unit of time, entering by infiltration on the windward side of the building, is assumed to escape at room temperature on the leeward side, the loss in heat from the room from this cause is given by the above formula.

The effect of water vapor mixed with air in this connection is so slight that it is neglected. For problems in which the humidity plays an important part the reader is referred to the chapter on "Air Conditioning."

The volume of air entering a building in this manner is not a matter of very precise calculation. It depends largely on the tightness or fit of the sash in the window frames, assuming that the windows and doors are kept closed, as has been proven by actual experiment. The use of weather stripping materially reduces this loss and is recommended for any installation.

Infiltration Based on Number of Air Changes. It has been the practice of engineers in the past to make an allowance for infiltration on the basis of a certain number of assumed air changes per hour based on the cubic contents of the room or building.

Let C = cubic contents of room or building, cu. ft.
N = assumed number of air changes per hour, measured at inside temperature.

TABLE 12

B.T.U. REQUIRED FOR HEATING AIR

This table gives the quantity of heat in British thermal units required to raise one cubic foot of air at initial temperature specified through a given temperature interval.

Initial Temperature Fahrenheit	Final Temperature Fahrenheit									
	40°	50°	60°	70°	80°	90°	100°	110°	120°	130°
−10°	1.051	1.262	1.473	1.684	1.892	2.102	2.311	2.522	2.732	2.943
0°	0.822	1.028	1.234	1.439	1.645	1.851	2.056	2.262	2.467	2.673
+10°	0.604	0.805	1.007	1.208	1.409	1.611	1.812	2.013	2.215	2.416
20°	0.393	0.590	0.787	0.984	1.181	1.378	1.575	2.771	1.968	2.165
30°	0.192	0.385	0.578	0.770	0.963	1.155	1.345	1.540	1.733	1.925
40°	0.000	0.188	0.376	0.564	0.752	0.940	1.128	1.316	1.504	1.692

Then $Q = NC$ cu. ft. of air entering by infiltration per hour.

The following estimated number of air changes per hour, as stated by *R. C. Carpenter*, for well-constructed buildings are frequently used in calculations.

TABLE 13

RESIDENCE HEATING

Halls	$n = 3$
Rooms on 1st floor	$n = 2$
Rooms on 2nd floor	$n = 1$
Offices and stores, 1st floor	$n = 2$ to 3
Offices and stores, 2nd floor	$n = 1\frac{1}{2}$ to 2
Churches and public assembly rooms	$n = \frac{3}{4}$ to 2
Large rooms with small exposure	$n = \frac{1}{2}$ to 1

Example. Required the heat loss, by infiltration, from a room containing 20,000 cu. ft., the temperature of which is maintained at 70° in zero weather, the estimated number of air changes being two per hour.

$Q = 2 \times 20{,}000 = 40{,}000$ cu. ft. of air entering per hour measured at 70°.

$h = 0.018 \times 40{,}000 \times (70 - 0) = 50{,}400$ B.t.u. per hour.

The above method, while simple to apply, can hardly be said to have a rational basis and frequently leads to rather absurd results.

Infiltration Based on Window Leakage.* As previously mentioned, by far the greater portion of the infiltration is due to loose fitting sash. This being the case, it would appear rational to base the infiltration loss on the amount of air which will leak into the room around the average window under the average wind movement during the coldest weather.

The experiments of *W. H. Whitten*, made in 1908, are of interest in this connection. These experiments were made on a double hung window 2′-0″ x 4′-0″, having an area of 8 sq. ft. and a perimeter including the meeting rail of 14′-0″. A test house (Fig. 10) was built and placed on the roof of a 25 story office building so that it turned on a pivot and faced the prevailing wind. The window was placed in the side facing the wind. The average width of opening between the sash and frame was 1/16″ for the "ordinary sash." This corresponds to a rather loosely fitted sash in practice. The following results were obtained:

TABLE 14

Velocity of Outside Air in Miles per Hour	Cu. Ft. per Minute Leakage Through 14′-0″ of Crack 1/16″ Clearance		Cu. Ft. per Min. per Ft. of 1/16″ Crack per Mile Veloc.
	Weather Stripped	Ordinary Sash	
6.0	1.0	12.	0.143
9.1	1.6	19.	.148
9.5	1.65	20.	.147
9.6	1.75	19.6	.148

* The reader is also referred to a comprehensive report on window leakage tests for both hollow metal and wood sash with and without weather strips in the Journal A. S. H. and V. E., Jan., 1916. These tests gave the following results corresponding to a wind velocity of 23 miles per hour:

Plain wood sash	1.9	cu. ft. of air per min. per ft. of perimeter.
" " " metal stripped	0.4	" " " " " " " "
Hollow metal sash	3.6 to 4.8	" " " " " " " "
" " " weather stripped	1.2 to 2.5	" " " " " " " "
Copper covered sash	2.2	" " " " " " " "

Under 6 miles per hour no perceptible leakage could be detected for the window fitted with the weather stripped sash (Fig. 11). It will be observed that the amount of air in cu. ft. per min. leaking in was approximately 2 times the wind velocity in miles per hour for the ordinary win-

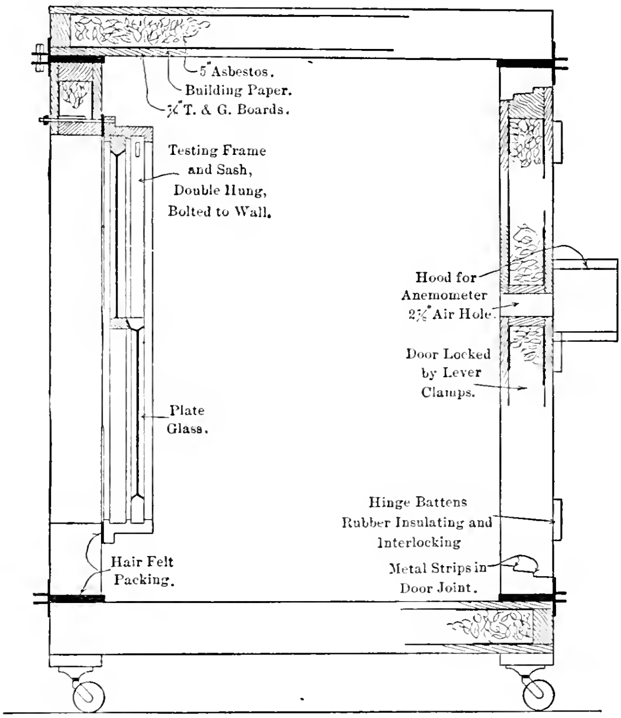


FIG. 10. BOX FOR AIR LEAKAGE TESTS THROUGH SASH.

dow, which corresponds to 0.147 cu. ft. per min. per ft. of crack per mile wind velocity and 0.013 cu. ft. per min. for weather stripped sash.

The following laboratory test to determine the air leakage around ordinary and weather stripped sash was made by *Ralph Collamore* in 1907 at Detroit, Michigan. The apparatus consisted of a motor-driven blower having a galvanized sheet steel cone fitted to receive the sash with one end connected with the blower and the opposite end arranged to receive an anemometer. The sash used consisted of two single lights, the total outside measurement being 2′ × 4′. The ordinary sash was fitted with 1/32″ clearance.

The results are given in Table 15.

Mr. R. P. Bolton in a paper read before the *Association of Edison Illuminating Companies* (1910) states that the most severe heating combination found from an examination of the records of New York city, covering a period of many years, was a wind velocity of approximately 28 miles per hour accompanied by a temperature of 20° F.

Mr. Whitten, in a paper read before the *American Society of Heating and Ventilating Engineers*, (1909) states that "the average wind movement in localities where heating is important is about 13 miles per hour during the heating season."

TABLE 15

Air Pressure Inches Water	Corresponding Wind Velocity Miles per Hour	Cubic Feet of Air per Minute Passing Around Sash	
		Ordinary Window	Weather Stripped Window
.05	10.78	15.0	....
.08	13.66	.....	1.4
.10	15.11	24.7	2.8
.20	21.61	37.4	5.3
.30	26.48	47.4	8.4
.40	30.57	57.4	9.6
.50	34.18	65.1	10.7
.60	37.44	73.1	11.7
.80	43.44	88.1	14.0
1.00	48.34	101.4	16.1

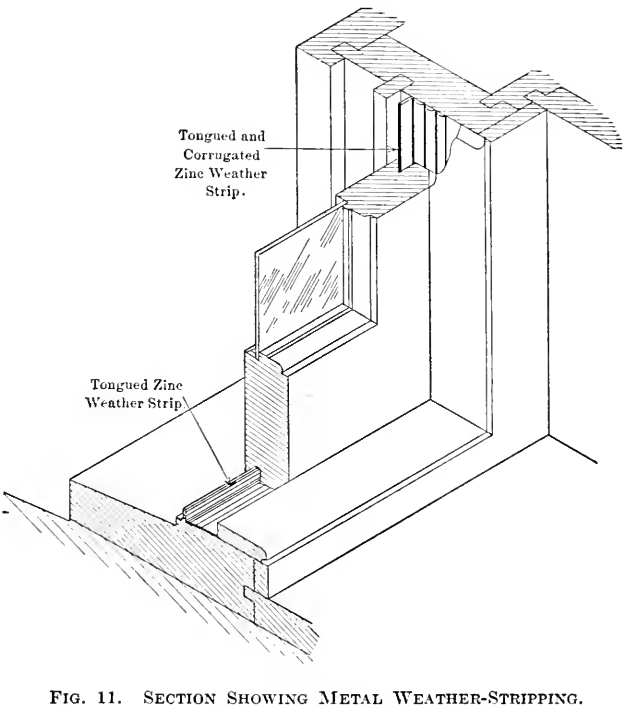


FIG. 11. SECTION SHOWING METAL WEATHER-STRIPPING.

Assuming the same ratio of air leakage as given by Table 14, the heat loss by infiltration would be, per degree difference between the inside and outside temperature, 60 × 13 × 0.147 × 0.087 × 0.24, or 2.4 B.t.u., per linear ft. of $^1/_{16}''$ crack per hour for an average wind movement of 13 miles per hour. The actual sash clearance determines the infiltration loss in any given case. In the general run of well constructed buildings this clearance should not exceed $^1/_{32}''$ while for poor construction the clearance may be as much as $^1/_{16}''$. By the above method the heat loss by infiltration may be approximately determined by the use of the following values:

TABLE 16

APPROXIMATE HEAT LOSS BY INFILTRATION

Construction	B.t.u. per Hr. per Ft. of Crack per 1° F.
Good construction $^1/_{32}$″ sash clearance	1.2
Poor construction $^1/_{16}$″ " "	2.4
Weather stripped sash	0.15

The infiltration of cold air takes place through the windward side and the warmed air leaves through the leeward side of the building. As any exposed wall may become the windward side, the infiltration, as calculated by this method, is determined by the *total lineal feet of sash and door clearance existing in the one outside wall having the greatest amount of glass and door surface,* and not upon the total number of feet of crack or clearance in all of the outside walls.

In any room the wall with maximum length of window crack must be used.

Infiltration Losses in High Buildings. The infiltration loss also increases with the height of the building since the wind pressure at an opening situated at a considerable height above the ground is greater than for an opening which is below the average level of buildings in the immediate neighborhood. This increase in wind velocity, according to *Mr. R. P. Bolton,* and its relative effect on infiltration are shown by the curve given in Fig. 12. It will be noted that no allowance is made for increased infiltration loss in buildings less than 100 feet in height.

Short Rules for Estimating the Heat Loss of Buildings. There are a great variety of "rule of thumb" methods for estimating the heat loss H for proportioning the heating surface required when direct radiation is to be used. These so-called practical rules are intended to be based on average building construction and ratio of wall and glass surface to the cubical contents as found in the class of building to which they refer.

These rules when modified for unusual conditions and applied by an engineer with long experience in the proportioning and design of heating systems produce satisfactory results. They are, however, rapidly being discarded except as used for a rough check on the more refined methods of calculation.

Carpenter's Rule. The following formula, or rule, which has been widely used for many years in this country, was proposed by *Professor R. C. Carpenter.* It is not intended to be applied to buildings covered with corrugated sheet steel or metal lath and plaster walls, unless the wall constant is changed to suit the condition.

By reference to Table 9, it will be noted that a fair average value for the heat transmission of the usual well constructed building wall is approximately 0.25 B.t.u. and for glass 1.0 B.t.u. per degree difference between the inside and outside temperature per hour.

Professor Carpenter states that usually we may neglect all inside walls, floors and ceilings and consider only the outside walls with sufficient accuracy.

The estimated number of air changes per hour, by infiltration, has already been given by Table 13.

Let C = Cubical contents of room in cu. ft.
n = Number of air changes per hour, see Table 13.
0.02 = B.t.u. to raise 1 cu. ft. of entering air 1° F.
W = Net wall surface in sq. ft.
G = Glass surface in sq. ft.
$(t - t_0)$ = Temperature difference between inside and outside.
H = Total heat to be supplied per hour in B.t.u.

$$H = (0.02\,n\,C + G + \tfrac{1}{4}W)\,(t - t_0).$$

The substantial accuracy of this formula was proven by several experiments conducted by

Professor Carpenter on actual buildings. (See *Trans. Am. Soc. of Heat. and Vent. Engrs., Vols. III* and *IV*.)

Mills' Rule. This rule was developed by *John H. Mills* about 1890, and gives the amount of direct steam radiating surface required to offset the heat loss. The rule is based on an outside and inside temperature difference of 70° F.

R = Direct steam radiating surface in sq. ft.

H, G, W, and C as for *Carpenter's Rule*.

$$R = \frac{C}{200} + \frac{W}{20} + \frac{G}{2}$$

As the average direct steam radiator emits approximately 250 B.t.u. per sq. ft. per hour, then $H = 250\ R = 1.25\ C + 12.5\ W + 125.\ G$.

For the amount of direct hot water radiating surface required add 60 per cent to the amount as calculated for steam.

Authors' Rule. The authors propose the following rule for application to the average well constructed building, which is similar to that of *Professor Carpenter's*, with the exception that

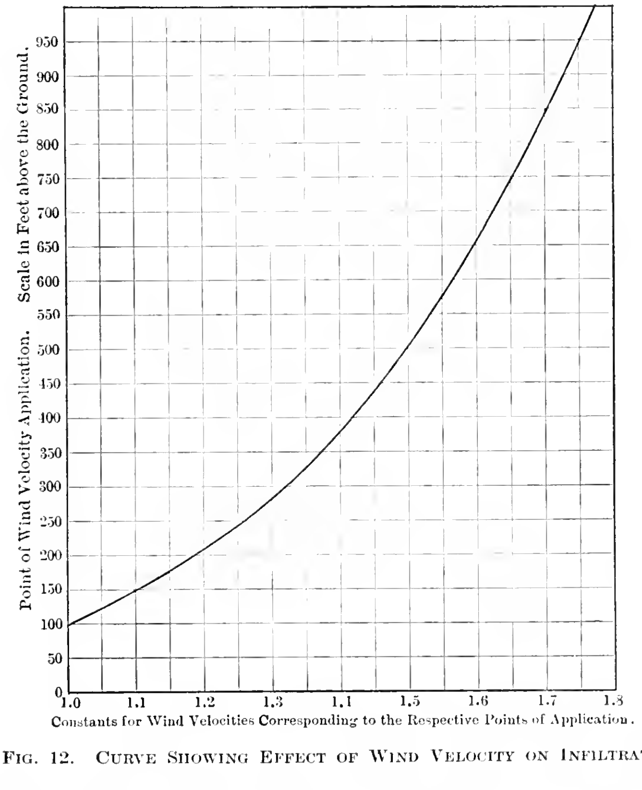


FIG. 12. CURVE SHOWING EFFECT OF WIND VELOCITY ON INFILTRATION.

the infiltration loss is based on the data given by Table 16, the notation used being the same as previously stated.

P = perimeter of windows in feet.

$H = (1.2\ P + \frac{1}{4}\ W + G)\ (t - t_0)$.

In the case of building walls constructed of corrugated sheet steel or metal lath and plaster the wall constant should be changed accordingly.

Estimating Boiler Horsepower and Coal Required for Heating. A simple method of estimating the boiler horsepower required for heating as well as the coal required per season has

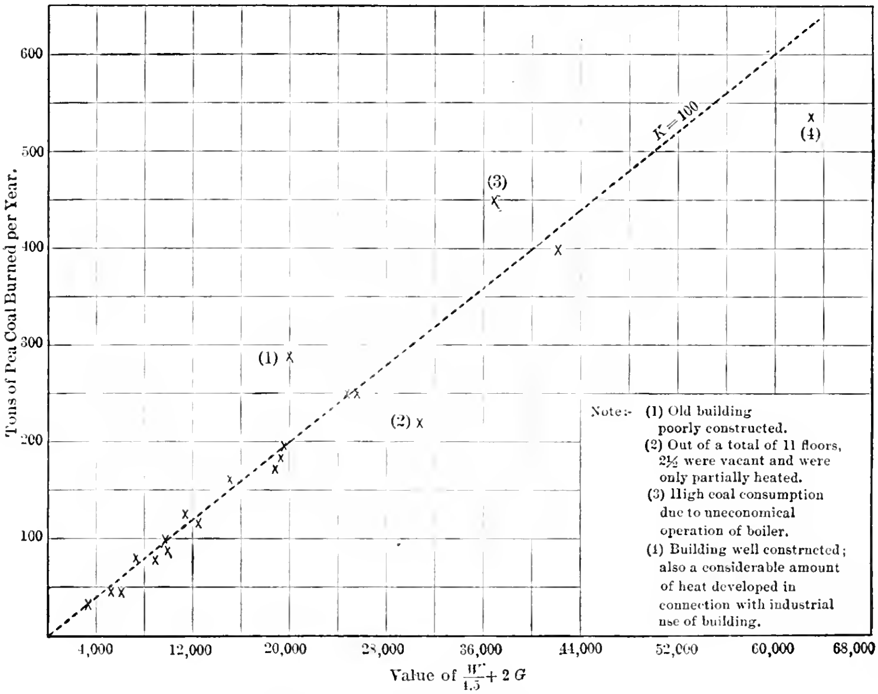


FIG. 13. RELATION BETWEEN COAL CONSUMPTION AND AREA OF EXPOSED SURFACES

been reported by *Mr. R. P. Bolton* in a paper on "The Heating Problem in Its Relation to Central Station Lighting and Power Service," as quoted in the following extract:

"The heat losses due to transmission and infiltration are first determined as follows:

(1) Transmission losses.

W = Exterior exposed wall surface in square feet $\times$ 21 = heat units per hour.

G = Area in square feet of window openings $\times$ 70 = heat units per hour.

(2) Leakage to be heated from all sides of building.

L = Perimeter of window openings in feet for ordinary construction $\times$ 72 = heat units per hour. Perimeter of window openings in feet for metallic stripped sashes $\times$ 31 = heat units per hour.

"The sum of all in heat units, plus 15 per cent for line and pipe losses, divided by 30,000 equals the boiler horsepower required per hour to meet the most extreme weather conditions.

"This total capacity placed on top of the diagram in Fig. 1 gives a scale by means of which all monthly hourly averages and totals may be ascertained.

"As the lineal feet in the perimeters of the window openings plus the meeting rail is, in standard types of windows, approximately equal to the amount of the area of the opening in square feet, the computation of wind leakage may be simplified by treating L as G, reducing the computation to

$$G \times 70 + G \times 72 = 142\ G$$

"The methods above outlined have been interestingly confirmed by observations conducted by *E. F. Tweedy*, of the *New York Edison Company*, the results of whose investigations are shown in Fig. 13.

"The actual consumptions of coal of a number of buildings were plotted and related to a basis of their cubical contents. It was observed that the plottings showed no definite relation to one another, and that the rate of coal consumption per season per 1000 cubic feet appears to be the same in the case of buildings widely different in total cubical contents.

"The same cases show a harmonious relation when plotted on a basis which takes into consideration the exposures of the buildings, with due allowance for air leakage.

"At the outset it was necessary to adopt an average ratio for the rate of heat transmission of glass to that of wall surface, and to obtain this it was, of course, necessary to adopt some thickness of wall as representing a fair average for loft and office-building construction in this city. Without entering into the details which led up to its adoption, a ratio of 4½ to 1 was finally adopted.

"Realizing the uncertainty which arises in providing for the leakage loss on the basis of assuming an arbitrary number of air changes per hour, it was decided to combine this loss with the transmission losses. Such a combination was deemed possible by the following line of reasoning: the leakage loss being proportional to the summation of the window perimeters (assuming the leakage space to be approximately constant in width) and the sizes of windows in similar types of buildings being fairly uniform, it becomes possible to consider the leakage loss as a function of the total glass surface without any very appreciable error.

"Having made the foregoing assumptions, the following equation is obtained:

$$T = C\left(\frac{W}{4.5} + G\right) + C'G$$

where

T = Tons of coal consumed per year.
W = Total exposed wall surface in square feet.
G = Total glass surface in square feet.
C and C' = Constants.

"As it is not necessary to solve the above equation, C and C' need not be actually determined; for purposes of plotting, however, a relation between C and C' must be obtained. For $C = C'$ the equation evidently becomes

$$T = C\left(\frac{W}{4.5} + 2G\right)$$

"The relation between C and C' can be varied considerably without affecting the results to any appreciable extent other than to alter the value of C. For $C = C'$, $C = 100$, which is a very convenient quantity for practical use. Fig. 13 shows the data for 20 office and loft buildings plotted from the equation in the above form. It will be seen that the values conform very closely to the equation as given. It will be observed that pea coal was burned in all of the buildings cited. Data were secured, however, from a number of buildings where the buckwheat coals or various mixtures were being used. All such cases—as would naturally be expected, due to the lower calorific value of such fuels—fall above the pea-coal line, but in most cases to a considerably greater extent than is to be explained by the difference in the thermal value of the fuels. This probably can be accounted for by the fact that these poorer coals are very rarely burned under proper conditions as regards type of grate and draft, thereby nullifying the saving which might otherwise result from the use of a cheaper fuel.

"For apartment-house buildings, the same general relation has been found to hold good, the only difference being that the value of the constant C is less, being about 60 instead of 100. This

can doubtless be explained by the generally poorer construction of such buildings and by the fact that heating is required for more hours during the day."

Examples of Heat Loss Calculations. It is convenient and advisable to adopt a standard form for recording the heat loss calculations. The method used in recording these data varies somewhat with the class of heating for which the calculations are made, as the form sheet usually contains the additional information as to the size of radiators, or hot air flues, registers, etc.

Examples in the calculation of the heat loss from buildings, and suggested methods for recording these data, are given later in the text under the chapters on "Direct Steam and Hot Water Heating" and also "Gravity Furnace Heating."

Heat Supplied by Persons, Lights, Motors, Machinery, Etc. (*c*) The quantity of heat emitted by persons is ordinarily not of sufficient importance to be taken into account, except in cases of assembly halls and theaters.

The following allowances may be made when required:

Persons

Man at rest . . . . 400 Bt.u. per hr.
Man at work . . . . 500 B.t.u. per hr.

The heat introduced by lights is as follows:

Lights

Electric Lamps. . . . B.t.u. per hr. equals watts per lamp × number of lamps × 3.415.
Gas lighting

1 cu. ft. producer gas . . . . 150 B.t.u.
1 cu. ft. illuminating gas . . . . 700 B.t.u.
1 cu. ft. natural gas . . . . 1000 B.t.u.

A Welsbach burner averages 3 cu. ft. of gas per hour and a fish tail burner 5 cu. ft. per hour.

Motors. Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

Machinery. If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used.

In the first case the B.t.u. supplied per hour $= \dfrac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2546$, and in the second case B.t.u. per hr. = d.h.p. × 2546, in which 2546 is the B.t.u. equivalent of 1 horsepower hour. In high-powered mills this is the chief source of heating and is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round

Equivalent Temperatures. The test for the fulfilment of a heating guarantee for certain specified inside and outside temperatures must often be undertaken when the latter temperature is other than that named in the guarantee. Under such conditions the corresponding inside temperature, equivalent to that guaranteed, must be computed and compared with the temperature as determined by test under the new conditions.

The relation between the equivalent and guaranteed inside temperature is found as follows:

Direct Radiation.

Let W = net wall area.
G = net glass area.
$Q = N \times C$ = volume of air leaking into room per hr. in cu. ft.
t and t_0 = guaranteed inside and outside temperatures.
t_s = guaranteed temperature of steam or water.
t' and t_0' = actual inside and outside temperatures under test conditions.
U_w and U_g = transmission coefficients for wall and glass.

K and K' = transmission coefficients for radiation under guaranteed and actual conditions.
R = sq. ft. of radiation installed.

Then the relation between the heat loss and the heat supplied by the radiation will be, as guaranteed, $(U_w W + U_g G + 0.02\,Q)\,(t - t_0) = K\,R\,(t_s - t)$ . . . . (1)

As found by test $(U_w W + U_g G + 0.02\,Q)\,(t' - t'_0) = K'\,R\,(t_s - t')$ . . . . (2)

If we divide equation (2) by equation (1) and assume $K = K'$, which is not quite true since K will always be greater than K' when the actual outside temperature is greater than the guaranteed outside temperature, and vice versa, we have:

$$\frac{t' - t_0'}{t - t_0} = \frac{t_s - t'}{t_s - t}, \text{ and on solving,}$$

$$t' = \frac{t_s\,(t_0' + t - t_0) - t_0' \times t}{t_s - t_0} \qquad (3)$$

Indirect Radiation. For an indirect system the relation is:

$$\frac{t' - t_0'}{t - t_0} = \frac{K\left[t_s - \dfrac{t'_1 + t_0'}{2}\right]}{K'\left[t_s - \dfrac{t_1 + t_0}{2}\right]} \qquad (4)$$

in which t_1' and t_1 are the temperatures of the air leaving the indirect radiator. In this case the values of K and K' will differ more than in the previous discussion, and it will be necessary to consider them equal and substitute t' and t for t'_1 and t_1, on the right hand side of the expression, in order to solve for t' as in equation (3). If this is done the expression for t' will be as given above.

The following table has been computed from equation (3) for steam at 5 lb. gage pressure and a guarantee based on zero temperature outside.

TABLE 17

EQUIVALENT INSIDE TEMPERATURES

Actual Outside Temperature Fahrenheit	Guaranteed Inside Temperature Fahrenheit			
	50	60	70	80
−10	42.2	52.6	63.1	73.5
0	50.0	60.0	70.0	80.0
+10	57.8	67.4	76.9	86.5
20	65.6	74.8	83.8	93.0
30	73.4	82.2	90.9	99.5

Equation (3) may be reduced to a much simpler form provided the terms of the guarantee are known, as for example, when $t_0 = 0°$, $t = 70°$ and $t_s = 227°$. Then, $t' = 0.691 \times t_0' + 70$, from which, values in the above table may be readily computed for the 70° column for other outside temperatures than those given.

CHAPTER IV

HEAT TRANSMISSION OF DIRECT RADIATORS

Heat Required and Supplied. The heat loss from buildings is usually supplied from metallic radiators of varying types, which may contain a variety of *heating media*, such as steam, water, air, or furnace gases. In order to determine how many square feet of radiation will be required to make up for any given heat loss, it is necessary to know the rate at which these radiators will emit heat, per sq. ft. of surface, *under such conditions as will exist in practice.* In the case of electric heating a metallic coil is heated by an electric current, due to the resistance offered to the flow of the current, and the heating effect produced per foot of length determines the amount of such coil required to furnish a certain number of B.t.u.

The radiators may be used as *direct radiators*, placed directly in the heated room, so that the heat is transmitted both by convection and radiation to the colder air and surroundings, or they may be so placed without the heated space as to warm the air entering the same by convection only, in which case they are known as *indirect radiators.*

Method in which Heat is Transmitted by Radiators. Whatever may be the heating media within the radiator, it gives up its heat by *contact* with the inner surface; this heat then passes through the walls of the radiator by *conduction*, and is finally given off from the outer surface by *radiation* and *convection* in direct heating, and practically by convection only in indirect heating. Radiant heat is transmitted directly *in straight lines* from the surface of the radiator to the object warmed, and is usually delivered at high temperature, while the convected heat is first absorbed by the air in passing over the radiator and is then delivered to the object warmed at a comparatively low temperature by a reverse process of convection, actual contact being always essential to heat transfer by convection.

The *transfer of heat through the metal wall* of the radiator involves, then, not only conduction of heat through the metal itself, but the exchange of heat between the heating media and the metal at the inner surface, and a discharge or emission of this heat from the outer surface to the surrounding air and objects. These surface transfers, especially the outer, are attended with more or less resistance to the passage of heat, due to the existence of a *surface film* of the media or air, which, in the latter case, is at a considerably higher temperature than the balance of the air passing over this film. The metal wall is therefore partially insulated and rendered less efficient than it would be if free from this film and in direct contact with the cooler outer air.

It will also be apparent that if the resistance at the inner surface is small and at the outer surface it is great the temperature of the metal wall will approach that of the heating media within, since the heat will be supplied to the inner surface much faster than it can be discharged by the outer surface. Furthermore the *thickness* of the metal wall, which is a good conductor, will be of little consequence, since the rate at which heat will pass will depend entirely on the rate with which it can be discharged from the outer surface.

Determination of Amount of Heat Transmitted. While it is entirely possible to investigate the emission of heat from radiators of definite geometrical proportions by a purely mathematical discussion, such as developed by *Peclet*, the actual radiators used today are of such irregular shapes that it is usually impracticable to apply the theoretical coefficients for radiation and convection as determined under ideal laboratory conditions.

For a given velocity V, the *total heat given off by radiation and convection* is expressed by $H = K(t_s - t_a)A$, where K is taken as a constant, known as the coefficient of transmission, t_s is the temperature of the heating media, t_a is the room temperature in the case of direct radiators and the average temperature in the case of indirect radiators of the air passing over the ra-

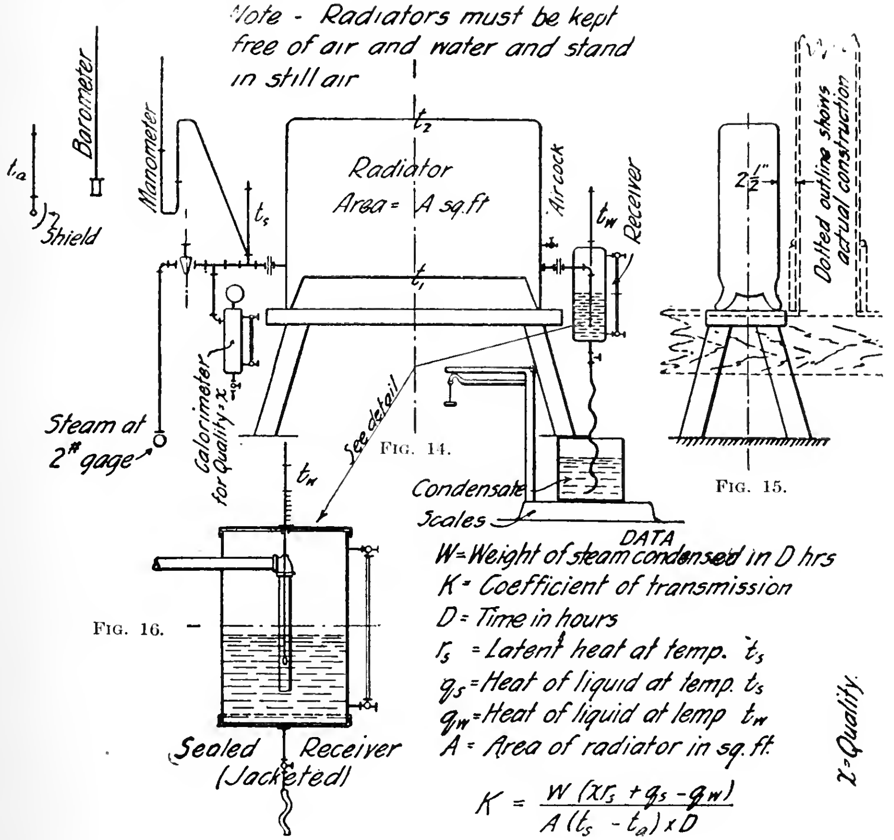


FIG. 14.

FIG. 15.

FIG. 16.

FIGS. 14, 15, AND 16.

diator, and A is the area of the radiator in sq. ft. K actually varies somewhat, being a function of the temperature difference $(t_s - t_a)$, as will be shown.

As a matter of fact less than half of the heat is given off by radiation in most direct radiators, and with indirect radiators the radiant heat is such a small proportion that it may be neglected. For an explanation of how the *radiation of heat is interfered with* in a direct radiator see Fig. 31 and note that no radiant heat can be delivered by the point "O" within the shaded areas.

Factors Affecting the Heat Transmission of Radiators. The amount of heat given off by radiation, per unit of surface per unit of time, is *independent* of the *form* and *extent* of the radiating surface, provided there are no re-entrant angles or surfaces to intercept the rays of radiant heat. It is *dependent* on the *nature* of the surface, *temperature difference* between surface and surroundings, and the value of the *absolute temperature* of the surface. Radiation increases much more rapidly than the temperature difference, and this increase is extremely rapid at high temperatures.

The amount of heat given off by convection, per unit of surface per unit of time, is *inde-*

pendent of the *nature* of the surface and its *absolute temperature*. It is *dependent* on the velocity of the air, probably varying as the $V^{\frac{1}{n}}$, on the *form* and *extent* of the surface, and on *difference* in *temperature* between the surface and the surrounding air.

The *practical significance of the above statements*, as will be seen by reference to Tables 18 and 20 and Figs. 29 to 32, is that single-column radiators are more efficient, *i.e.*, have a higher coefficient of transmission than multiple-column radiators. Also low direct radiators or narrow indirect are more efficient than high direct, or long indirect radiators, in which latter cases part of the surface is surrounded by air already heated to a high temperature. Also radiators of dull and dark finish are more efficient than those with bright or polished surfaces, such a surface as lampblack or soot being most efficient for radiating heat. Finally, the colder the surroundings and the air in contact with the radiator, or the hotter the heating media, the more efficient each sq. ft. becomes.

Heat-Transmission Tests. The actual *determination of the coefficients of transmission* for radiators using steam or water as the heating medium is usually made by running a condensation or heat-transmission test on the exact type of radiator to be investigated, under conditions simulating those which will exist when the radiator is installed for heating a building. *The coefficient of transmission is the number of B.t.u. given off per sq. ft. of the radiator surface per hour per degree difference in temperature between the heating media and the surrounding air.* No standard method or code for testing radiators has been adopted, and the tests already made by *Allen, Carpenter, Barrus, Monroe,* and others have been conducted according to their judgment.

It is most important that the interior of the radiator be kept free from air and water, and that the radiator shall stand in perfectly still air except for the natural convection currents.

One of the most important preliminary details is *the measurement of the heating surface,* as the coefficient obtained from any test is directly dependent on the amount of surface credited to the radiator under test. The methods employed for arriving at this surface usually consist in cutting out small patterns of paper, cardboard, or tinfoil, and fitting these as accurately as possible over each small irregularity, until a complete set has been secured for each typical section of the radiator. These small areas are then planimetered and totaled, and this total multiplied by the number of sections in the radiator, to get the area in sq. ft. This area is known as the *actual* surface, and may be larger or smaller than the *rated* surface, as given by the manufacturers.

A method of testing direct steam radiators as carried out by the authors at the *University of Illinois* is shown in Figs. 14 to 16, and details of the receiver are also given.

Sample Test for determining K, the coefficient of transmission, for a cast-iron radiator 3 columns wide, 10 sections long, and 38″ high.

Observed Data—

Pressures:

Barometer = 29.3″ Hg. = 14.37 lb. or 29.3 × 0.491.
Manometer 2.1 lb. = 2.10 " gage.
16.47 " absolute.

Temperatures:

Steam at 16.47 lb. = 217.8° F. Quality = x = 98 per cent,
Air = 73.2° F. by calorimeter.
Condensation leaving radiator = 216.3° F.

Weights:

Final = 116.5 lb.
Initial = 68.7 "
Net = 47.8 " condensation in 3.7 hours.

Areas:

Per section = 5.07 sq. ft. (actual).
10 sections = 50.7 " "

Time:

Elapsed = 3.7 hrs.

Calculations—

Coefficient

$$K = \frac{W \times (q_s + x\,r_s - q_w)}{(t_s - t_a) \times A \times T}$$

$$K = \frac{47.8 \times (186 + .98 \times 966.6 - 184.4)}{50.7\,(217.8 - 73.2) \times 3.7}$$

$$K = \frac{47.8 \times 948.6}{27{,}130}$$

K = 1.67 B.t.u. per hr., per 1° F., per sq. ft.

$$\text{Weight of steam} = \frac{47.8}{50.7 \times 3.7} = 0.255 \text{ lb. per sq. ft. per hr.}$$

$$\text{Density of steam} = \frac{1}{v} = \frac{1}{24} = 0.0417 \text{ lb. per cu. ft.}$$

Factors Affecting the Heat Transmission of Radiators. The heat transmission of direct radiators or the *radiation factor* in B.t.u. per sq. ft. per hour is $H = K\,(t_s - t_a)$, and it is seen that H will increase as t_s, the temperature of the heating media, increases, or t_a, the temperature of the room air, decreases. Also it should be specially noted that K, the coefficient of transmission, is *not* a *constant* even for the same radiator (Table 18), since K increases as $(t_s - t_a)$ grows large.

TABLE 18

TESTS BY *JOHN R. ALLEN* TO SHOW THE EFFECT OF VARYING DIFFERENCES IN TEMPERATURE BETWEEN THE HEATING MEDIUM WITHIN AND THE AIR SURROUNDING THE RADIATOR

Difference in Temperature Steam to Air	B.t.u. Transmitted per 1° Difference, per Sq. Ft. per Hour = K
80	1.425
90	1.455
100	1.485
110	1.515
120	1.550
130	1.590
140	1.635
150 Usual range in direct steam heating	1.665
160	1.710
170	1.745
180	1.770
190	1.815

NOTE—A 38″ high, 2-column radiator was used.

In general heating practice it is customary to neglect the variation in the value of K, since the temperature difference is usually about 140° to 150° F. For unusual temperature ranges the corresponding value of K should be used.

The *effect of painting* a radiator, as already intimated, is to influence the loss of heat from the surface only, and depends largely on the radiation factor for the surface coating in question; apparently the convection factor is but slightly affected by the surface coating.

TABLE 19

Tests by *Allen* and *Carpenter* on the effect of painting do not agree very closely; the former gives the relative transmission as:

Bare iron	= 1.00
Aluminum and copper-bronze	= 0.75
Snow-white enamel	= 1.01
White lead paint	= 0.987
White zinc paint	= 1.01

NOTE—The number of coats has very little effect on the transmission, which seems to depend entirely on the last coat applied.

Coefficients of Transmission for Direct Steam Radiators. The following table by the authors is based on the average performance of direct steam radiators standing exposed in air at 70° with steam at 220°, with a standard temperature difference of 150°:

TABLE 20

VALUES OF K FOR DIRECT RADIATORS

Type of Radiator	Height of Radiator			
	20 and 22 in.	26 in.	32 in.	38 in.
1 column	1.95	1.90	1.85	1.80
2 "	1.80	1.75	1.70	1.65
3 "	1.70	1.65	1.60	1.55
4 "	1.60	1.55	1.50	1.45
Flue, 42 sq. ft.	....	....	....	1.57*
Window	1.85	....	....	....
Pipe coils	2.00	....	....	....
Wall (horizontal)	1.95	....	....	....
" (vertical)	1.90	....	....	....

* Air entering flues at 70° and leaving same at 152°.—*Allen.*
NOTE—K increases (1) as height of radiator is reduced, and (2) as number of columns or width of radiator decreases.

In order to apply the coefficients given in the preceding table to conditions other than standard, it is only necessary to know the variation in K for a given increase or decrease in the temperature range above or below 150°, the standard range. An examination of test data so far available such as given in Table 18 seems to indicate, and has been so reported by *Chas. A. Fuller*, that this variation can be expressed as very nearly 0.2 per cent per degree above or below the standard range of 150°.

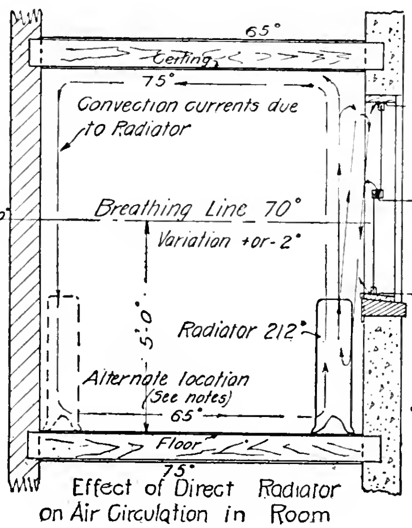


FIG. 17

Thus, if a 3-column, 38″ high, direct radiator is to be used in a room kept at 60°, with steam at 230°, we would have a temperature range of 170°, or 20° above standard, and the value of K would become $(1.55 + 0.002 \times 20 \times 1.55) = 1.61$, and each sq. ft. of radiation would give off $1.61 \times 170 = 274$ B.t.u. per hour.

Coefficients of Transmission for Direct Hot-Water Radiators. Table 20 preceding may be used for values of K for hot-water radiators of the same type as there listed, but allowance should be made for the lower temperature range in hot-water heating. Thus with a room usually at 70°, and water at 180° entering, and at 160° leaving the radiator, the temperature range is only 100°, or 50° less than the standard range. Then for a 2-column 26″ high direct radiator the value of K becomes $(1.75 - 0.002 \times 50 \times 1.75) = 1.58$, and each sq. ft. of this radiation would give off, $1.58 \times 100 = 158$ B.t.u. per hr.

The *effect of a direct radiator on* the *temperature* and the *air movement* within a room depends upon its location. If placed under a window (Fig. 17) there will be a general room circulation as shown by arrows, with a local circulation at the cold glass surface in the reverse direction if the outside temperatures are very low. By this arrangement the cold outside wall and win-

dow surface is in direct contact with the hottest air, and in close proximity to the hot surface of the radiator, resulting in a maximum heat transmission or loss through the wall, and corresponding load on the radiator.

Placing the radiator in the alternate position, shown dotted, and against the inside wall, will result in a reversal of air flow from that shown with no conflicting currents at the window, and a lower average temperature of air flowing down over the outside wall and window with the hot surface of the radiator far removed from this cold surface. This will result in a smaller heat-transmission loss than in the first case, and therefore means a smaller load on the radiator for the same breathing line temperature of 70° F. The variation in the air temperatures at the floor and ceiling from each other and from the mean temperature will be greater in the second case than in the first. This is due to the fact that cold air from the outside walls will be drawn directly across the floor to the radiator placed on an interior wall, and after being heated will be discharged to the ceiling at a higher temperature than in the first case.

The Effect of Enclosures on the Heat Transmission of Concealed Radiators. The use of *concealed* or *screened radiators* is becoming more and more common in the equipment of many classes of modern buildings. As a result of this concealment, placed around or above the radiator, the heating efficiency is generally reduced as compared with the same radiator when entirely exposed. Suitable *allowance* must therefore be made for this decrease in the value of K and *larger* radiators must be used whenever concealment is employed.

The *effect of enclosures* on the heat-transmitting power of radiators has been very completely investigated by *Professor K. Brabbée*, of the *Royal Technical Institute of Berlin*, and certain of his results and conclusions are here given as translated by *George Stumpf, Jr.*

These tests were made on plain surface cast-iron radiators of 10 sections each, spaced 3″ on centers. The 2-column radiators were 8½″ wide and the 3-column radiators were 9″ wide. Preliminary tests (Table 21) were run with each of the radiators in order to determine the coefficient of radiation K when the radiator was not enclosed, but standing in still air and placed 2½″ from the wall. This coefficient is the number of B.t.u. transmitted per hour, per sq. ft., per 1° F. difference between the mean temperature of the heating medium and the surrounding air.

TABLE 21

VALUES OF K FOR RADIATORS NOT CONCEALED

Height in Inches	No. of Columns	Value of K
49	2	1.62
50	3	1.38
24	2	1.74
26	3	1.50
The following values were obtained by *J. R. Allen* and compare favorably with the above		
38	2	1.65
38	3	1.45

When the 24″ 2-column radiator, 30″ in length and containing 10 sections, was placed 6″ from a wall having a cold exposure, the value of K was reduced 3.5 per cent as compared with the same radiator placed only 2½″ from the same wall.

The effect of enclosures as *increasing* or *decreasing* the value of K will be expressed as P and given in percentage, " + " or " − " respectively. The *free area* of screens, grilles, or registers will be expressed as A in percentage of total area. In this way the effect of an enclosure will be correctly expressed since the percentage will remain constant even though K may vary for the same radiator due to exposed locations or an unusual temperature range. In all cases I and O represent inlet and outlet respectively.

Case I

The *most efficient spacing for an enclosed radiator* (Fig. 18) was found to be 2½″ at front and rear. Increasing this space caused a greater decrease in the transmission efficiency than reducing the space.

It was also shown that for the same free area the design of the grille has no appreciable effect on the transmission efficiency of the radiator.

Case II

The *effect of enclosures without a top* (Fig. 19) was to *increase* the transmission efficiency of the radiator by about 12 per cent when the area of the inlet in sq. in. was approximately 10 times the heating surface in sq. ft. It will be seen that the air inlet should be made large, but that extending the height of enclosure had very little effect on increasing the efficiency.

For *maximum efficiency* the air must travel over the enclosed radiators with as *high a velocity* as possible. This *velocity must be uniform* over the entire length of the radiator.

Case III

The *effect of shelves at various heights above the top* of radiators (Fig. 20) is to reduce the transmission efficiency at least 2.5 per cent for narrow shelves and 5 per cent for wide shelves which are not over 1½″ above the radiator, for high radiators, 32″ and above. In the case of low radiators of 26″ height and less, this reduction will amount to 5 per cent for shelves 3″ above and as much as 10 per cent if placed only 1½″ above the radiator.

The tests were made with *deflectors* under the shelves, but it was found that these devices had but very little, if any, beneficial effect.

Case IV

The *effect of an open recess*, with the front of radiator flush with the face of wall (Fig. 21) is to *reduce* the transmission efficiency depending on space, R back of radiator, and distance a above radiator. The space S at ends of radiator has practically no effect on the transmission efficiency.

Actual tests on a recessed radiator show that the space R should be made 2½″ for maximum efficiency, and that with a distance of $a = 3''$, the minimum allowable, a reduction of 8 per cent in the heat-transmission efficiency will have to be made.

Case V (*a*)

The *effect of enclosures with various sizes of outlet screens at the top* of the enclosure (Fig. 22) is to *reduce* the transmission efficiency of the radiator. This reduction amounts to about 15 per cent when the screen is at the top of the enclosure and has the same length and width as the radiator, while the inlet is an open slot 2″ high and same length as the radiator. Screens wider than the radiator have but a slight advantage in improving the heat transmission, but for every inch the screen is made narrower about 5 per cent more should be deducted from the transmission coefficient. Free area of all screens used in tests was 44 per cent of total area.

Case V (*b*)

The *effect of enclosures with various sizes of open inlet slots* (Fig. 22) on high radiators 32″ and above is less marked than in the case of low radiators. With high radiators and slots from 3″ to 4″ high the *reduction* in transmission efficiency will not exceed 15 per cent, while with low radiators, 26″ and below, this reduction for a slot 2″ high, which is much used, will amount to about 25 per cent.

Case V (*c*)

The *effect of various sizes of screened inlets for enclosures* (Fig. 22) is to *reduce* the transmission efficiency of high and low radiators by about the same percentage as for open slots, provided the free area of the screen is made equal to the open slot area. For example, a screened inlet

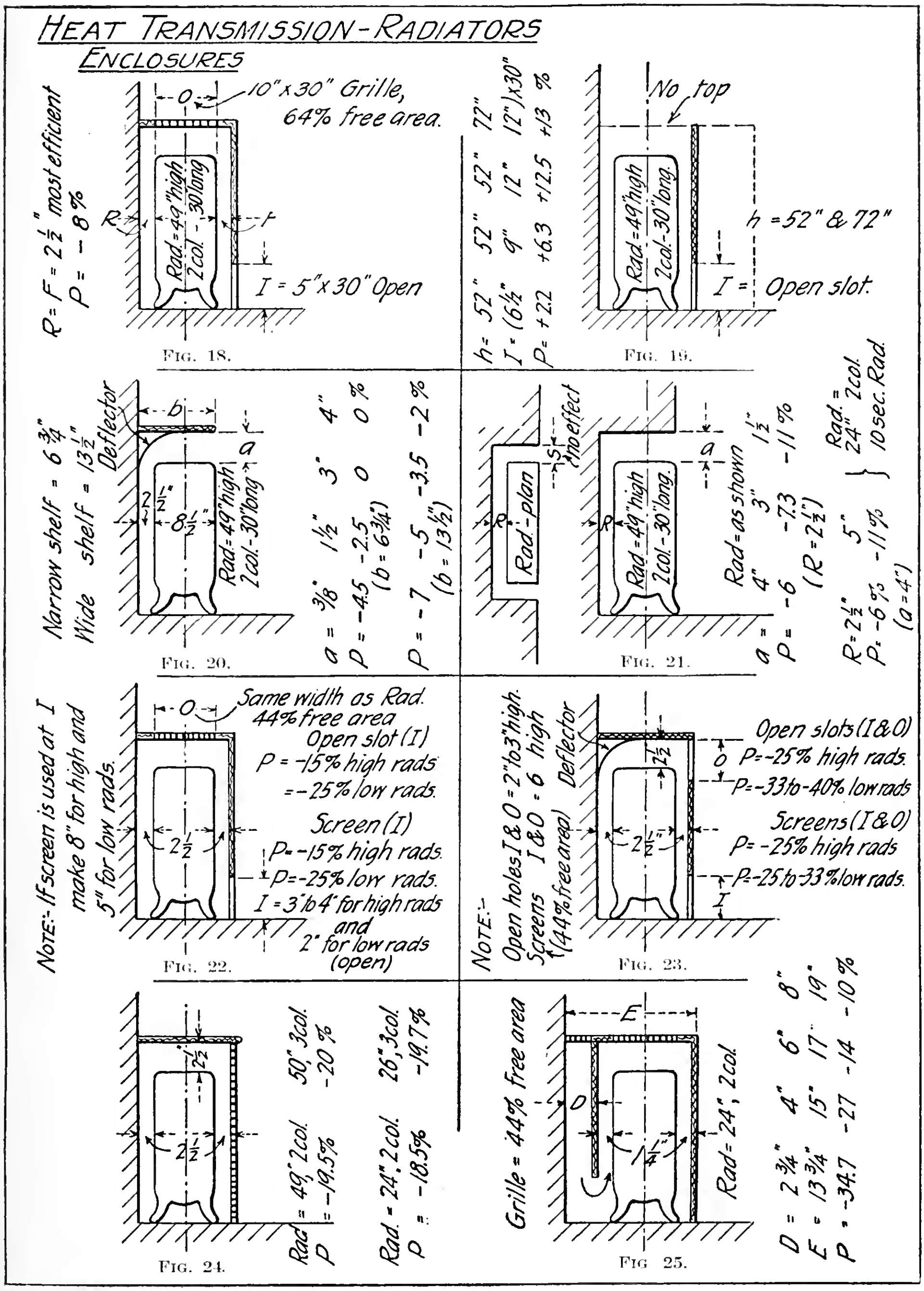


FIG. 18. FIG. 19. FIG. 20. FIG. 21. FIG. 22. FIG. 23. FIG. 24. FIG. 25.

8″ high by the length of the radiator has an equivalent free area equal to a slot 3½″ high, and the proper allowance for this enclosure would be to deduct 15 per cent for a high radiator.

In the case of low radiators, however, the loss of efficiency due to enclosures with screened inlets is about 25 per cent, as already given for open slots of the same area as the free area (taken at 44 per cent) of the screen.

Herewith is a sample test covering the above case: Radiator = 24″ high, 2 columns, 10 sections, 30″ long. Air inlet screened = 5″ × 30″, A_i = 44 per cent. Equivalent open slot = 2¼″ × 30″. Air outlet, also screened = 8½″ × 30″. A_o = 44 per cent. P = − 26.5 per cent.

All outlet screens should be at least as wide as radiator and the same length as radiator.

Case VI (*a*)

The *effect of enclosures with open slots for air inlet and outlet in front of enclosure* (Fig. 23) when top of enclosure is 2½″ above the radiator, is to *reduce* the transmission efficiency about 25 per cent for high radiators and about 33 per cent to 40 per cent for low radiators, the smaller value applying to low 3-column and the latter to low 2-column radiators. The slots were between 2″ and 3″ high, and as long as the radiator.

Case VI (*b*)

The *effect of enclosures with air inlets and outlets screened* and placed in front of enclosure (Fig. 23) is to *reduce* the efficiency for screens 6″ high and as long as the radiator by 25 per cent for high radiators and by 25 per cent to 33 per cent for low radiators of 2 and 3 columns respectively.

Moreover, if conditions require the inlet and outlet screens to be reduced to 4″ in height, the reduction in efficiency will amount to 40 per cent. The free area A of the screens is 44 per cent. The *deflector* appears to lessen the reductions as given above by about 2.5 per cent to 5 per cent.

Case VII

The *effect of enclosures with a perforated sheet-steel front* (Fig. 24) with top of enclosure 2½″ above the radiator is to *reduce* the transmission efficiency of 2- and 3-column low and high raditors by about 20 per cent.

Case VIII

The *effect of enclosures in window recesses or seats with an air supply taken through top of enclosure* (Fig. 25) is to *reduce* the transmission by from 30 per cent to 40 per cent, the latter applying to low radiators. In this arrangement it is seldom possible to make the space D more than 3″ to 4″ wide.

Tests with hot-water radiators not enclosed and placed 2½″ from the wall were made under the following conditions: flow temperature 176° F., return temperature 140° F., and average temperature 158° F., with surrounding air at 68° F. The results showed a variation of 10 per cent in value of K in favor of narrow radiators as compared with wide radiators of the same height, and a similar variation of 10 per cent in value of K in favor of low radiators as compared with high radiators of the same width.

Tests of *hot-water radiators in enclosures* agreed so nearly with tests of steam radiators similarly placed that values of P as expressed in percentage for steam radiators are directly applicable to water.

The *application of values of P* to practice can only be made when values of K for radiators standing 2½″ from the wall in still air, without enclosures, are known. These coefficients as determined by *Rietschel, Brabbée, Allen,* and other well-known authorities are given in Table 20.

Example. Assume that a certain room has a heat loss, including transmission through wall and glass and air leakage, of 10,000 B.t.u. per hour, and is to be heated by one steam radiator 20″ high, of 4 columns, located in an enclosure such as shown in Fig. 22. The air inlet is an open slot 2″ high and the outlet is a grille or screen of the same width and length as the radiator with a free area of 44

per cent of the gross area. By reference to Fig. 22 and Case V (*b*) it will be seen that the value P is -25 per cent. Also the value K for such a radiator when not enclosed is 1.60 B.t.u., and with steam at 219° F. the total emission per sq. ft. is 240 B.t.u. See Table 20.

But since this radiator suffers a reduction of 25 per cent in transmission efficiency due to the enclosure, the actual transmission is only 180 B.t.u. per sq. ft. Hence a radiator of $10{,}000/180 = 56$ sq. ft. is required.

The *pipe size* must be based on a radiator of 42 sq. ft., transmitting 240 B.t.u. or on a 56 sq. ft. radiator giving off 180 B.t.u. per sq. ft. It is best to reduce to B.t.u. in order to avoid confusion in selecting pipe size.

The *heat transmission of indirect radiators* for gravity and mechanical or fan blast systems will be discussed under those systems.

RADIATORS FOR DIRECT HEATING

Essential Features. Radiators for *direct steam and water heating* are constructed of *cast iron, wrought iron* or *steel.* The cast-iron radiator is usually built up of *sections* which are

NIPPLE CONNECTIONS FOR DIRECT RADIATORS.
U. S. Radiator Corporation.
Average dimensions are shown.

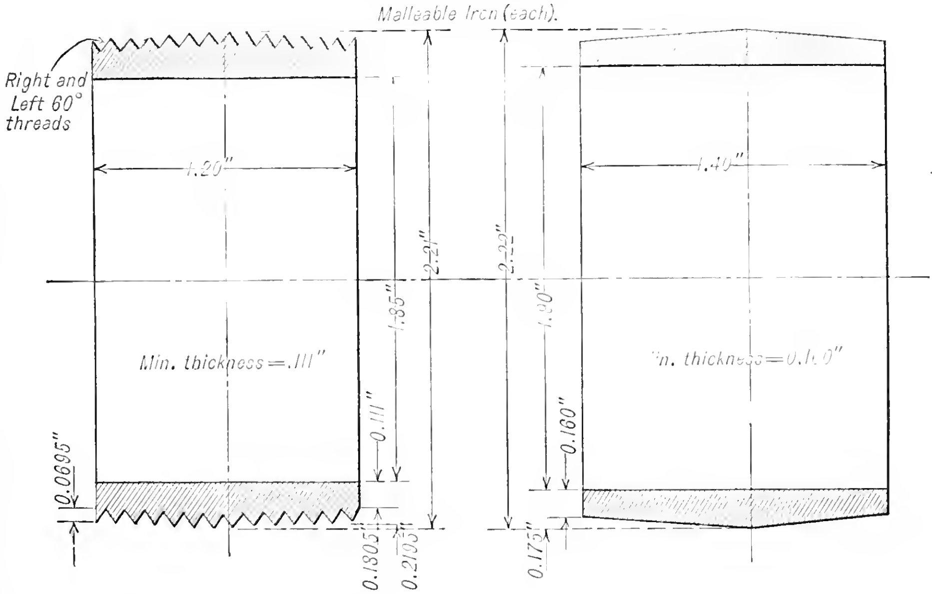


FIG. 26. MALLEABLE IRON SCREW NIPPLE. FIG. 27. MALLEABLE IRON PUSH NIPPLE.

put together with *screw* or *push nipples* (Figs. 26, 27, and 28). The sections are known as either *leg sections* or *loop sections,* the latter being the intermediate sections without feet.

Nipples. The *screw nipples* are of malleable cast iron with a 60° right and left thread (Fig. 26) and for water are 1½", and for steam 2" pipe size. Four-column water radiators have 2" nipples. The 60° thread has replaced the old 90° thread, which required a paper gasket between the *hubs* of each pair of sections (Fig. 28). With this new thread it is possible to make the joint "metal to metal" on the thread itself instead of at the hub.

Each nipple has two heavy projecting *lugs* cast on the inside of same so that an ordinary

piece of bar iron, flattened at one end for the length of the nipple, can be inserted to any point, and by applying a wrench to bar the nipple can be unscrewed, independently of all other nipples. These *nipple wrenches* can be furnished by the radiator manufacturers.

The *push* or *slip* nipples (Figs. 27 and 28) are made of steel or malleable iron and ground slightly tapered so that when inserted in the hub, which has a corresponding taper, the sections

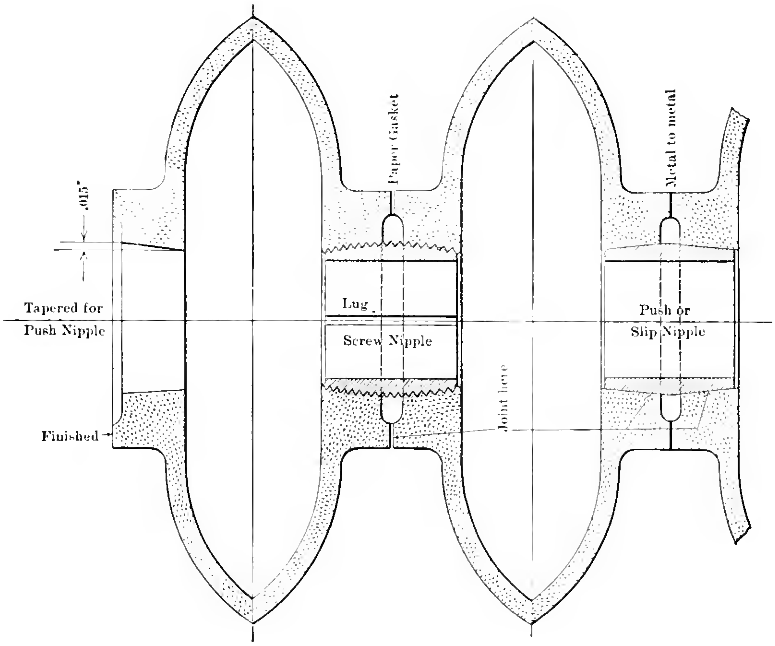


FIG. 28. SECTION THROUGH RADIATOR HUBS SHOWING METHOD OF MAKING JOINTS.

may be bound together by through rods and the nipples forced to a tight " metal to metal " fit in their conical seats.

The *malleable iron push nipple* (Fig. 27) is to be preferred to the steel push nipple, which is usually much lighter, and far more liable to corrosion.

Column Radiators. The *column radiator* (Figs. 33 to 36) is probably the most common type of radiator in use to-day, and the sections of which it is composed may be of one, two, three, or four *columns* (Figs. 29 to 32). These radiators may be of plain or ornamental design, but the former are certainly the more sanitary. They may be made up in a variety of *shapes* of one-, two-, and three-column radiators, such as *circular, corner, curved, ventilating, stairway, hot closet, marble topped,* and may be *legless* with *loop* sections only, supported on *concealed brackets,* or they may have *extra high legs.* (See *Special Radiators.*)

The data and dimensions ordinarily listed by the manufacturers of direct column radiation are given in Tables 22, 23, and 24 for a plain surface radiator known as the Peerless. The table of *heating surface* has been made out to 10 sections only, whereas the manufacturer's catalog lists radiators up to 32 sections. For radiators of a greater number of sections than 10 simply take multiples or combinations of the values here given or refer to the maker's catalogs. The Peerless radiator is so nearly standard that it can well be used for reference in laying out any work where column radiation is to be used.

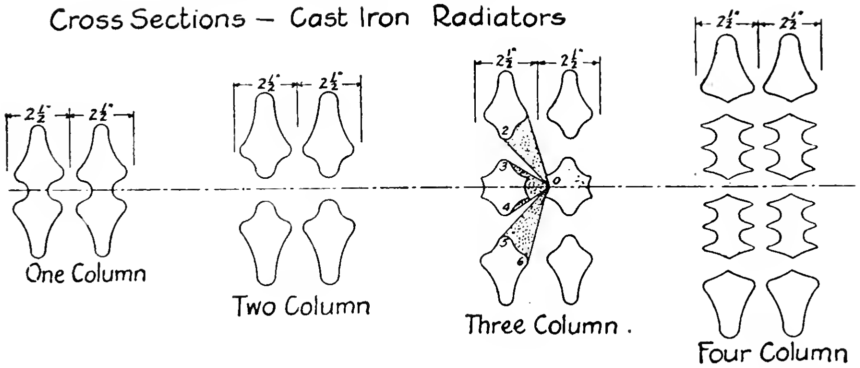


FIG. 29. FIG. 30. FIG. 31. FIG. 32.

TABLE 22

PEERLESS RADIATORS

(See Figs. 33 to 36.)

For Steam and Water *American Radiator Co.*

No. of Sec-tions	* Length 2½ In. per Section	HEATING SURFACE—SQUARE FEET								
		45-In. Height	38-In. Height	32-In. Height	26-In. Height	23-In. Height	22-In. Height	20-In. Height	18-In. Height	†15-In. Height
SINGLE COLUMN										
1	2½		3 Sq. Ft. per Section	2½ Sq. Ft. per Section	2 Sq. Ft. per Section	1⅔ Sq. Ft. per Section		1½ Sq. Ft. per Section		
2	5	....	6	5	4	3⅓	....	3	....	....
3	7½	....	9	7½	6	5	....	4½	....	....
4	10	....	12	10	8	6⅔	....	6	....	....
5	12½	....	15	12½	10	8⅓	....	7½	....	....
6	15	....	18	15	12	10	....	9	....	....
7	17½	....	21	17½	14	11⅔	....	10½	....	....
8	20	....	24	20	16	13⅓	....	12	....	....
9	22½	....	27	22½	18	15	....	13½	....	....
10	25	....	30	25	20	16⅔	....	15	....	....
TWO COLUMN										
1	2½	5 Sq. Ft. per Section	4 Sq. Ft. per Section	3⅓ Sq. Ft. per Section	2⅔ Sq. Ft. per Section	2⅓ Sq. Ft. per Section		2 Sq. Ft. per Section		1½ Sq. Ft. per Section
2	5	10	8	6⅔	5⅓	4⅔	....	4	....	3
3	7½	15	12	10	8	7	....	6	....	4½
4	10	20	16	13⅓	10⅔	9⅓	....	8	....	6
5	12½	25	20	16⅔	13⅓	11⅔	....	10	....	7½
6	15	30	24	20	16	14	....	12	....	9
7	17½	35	28	23⅓	18⅔	16⅓	....	14	....	10½
8	20	40	32	26⅔	21⅓	18⅔	....	16	....	12
9	22½	45	36	30	24	21	....	18	....	13½
10	25	50	40	33⅓	26⅔	23⅓	....	20	....	15

* Add ½ inch to length for each bushing.

† 15-inch height is made for steam only.

PEERLESS COLUMN RADIATORS FOR STEAM AND WATER.

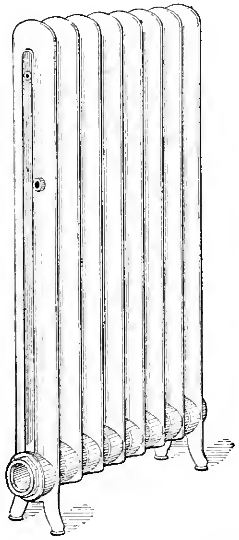


FIG. 33. SINGLE COLUMN

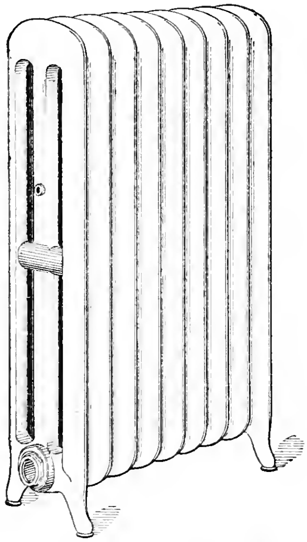


FIG. 35. THREE-COLUMN.

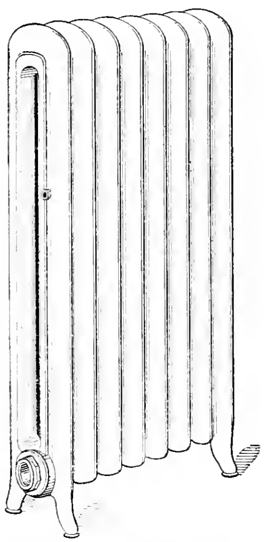


FIG. 34. TWO-COLUMN.

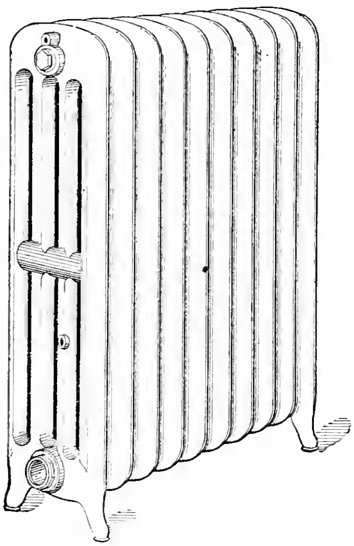


FIG. 36. FOUR-COLUMN.

TABLE 22—*Continued.*

No. of Sections	*Length 2½ In. per Section	Heating Surface—Square Feet								
		45-In. Height	38-In. Height	32-In. Height	26-In. Height	23-In. Height	22-In. Height	20-In. Height	18-In. Height	†15-In. Height
THREE COLUMN										
1	2½	6 Sq. Ft. per Section	5 Sq. Ft. per Section	4½ Sq. Ft. per Section	3¾ Sq. Ft. per Section		3 Sq. Ft. per Section		2¼ Sq. Ft. per Section	
2	5	12	10	9	7½	....	6	....	4½	....
3	7½	18	15	13½	11¼	....	9	....	6¾	....
4	10	24	20	18	15	....	12	....	9	....
5	12½	30	25	22½	18¾	....	15	....	11¼	....
6	15	36	30	27	22½	....	18	....	13½	....
7	17½	42	35	31½	26¼	....	21	....	15¾	....
8	20	48	40	36	30	....	24	....	18	....
9	22½	54	45	40½	33¾	....	27	....	20¼	....
10	25	60	50	45	37½	....	30	....	22½	....
FOUR COLUMN										
1	3 In. per Section	10 Sq. Ft. per Section	8 Sq. Ft. per Section	6½ Sq. Ft. per Section	5 Sq. Ft. per Section		4 Sq. Ft. per Section		3 Sq. Ft. per Section	
2	6	20	16	13	10	....	8	....	6	....
3	9	30	24	19½	15	....	12	....	9	....
4	12	40	32	26	20	....	16	....	12	....
5	15	50	40	32½	25	....	20	....	15	....
6	18	60	48	39	30	....	24	....	18	....
7	21	70	56	45½	35	....	28	....	21	....
8	24	80	64	52	40	....	32	....	24	....
9	27	90	72	58½	45	....	36	....	27	....
10	30	100	80	65	50	....	40	....	30	....

WIDTHS OF PEERLESS COLUMN RADIATORS

Type	Width of Section	Width of Legs
One-column	4½″	5½″
Two-column	7⅜″	8½″
Three-column	9″	10″
Four-column	10½″	11¼″

NOTES

Tappings: 2 inches and bushed as specified.

Connections: (For 1-, 2-, and 3-column radiators). Water—extra-heavy right and left threaded nipples at top and bottom. Steam—with extra-heavy right and left threaded nipples at bottom only.

Connections: (For 4-column radiators). Both steam and water—extra-heavy right and left threaded nipples at top and bottom.

Low-Drip Hubs: (For 1-, 2-, and 3-column radiators). One pipe steam—on supply leg section; two pipe steam—on return leg section.

Measurements: Center of tappings to floors, and between centers of upper and lower water tappings, see Tables 23 and 24.

* Add ½ inch to length for each bushing.

† 15-inch height is made for steam only.

TABLE 23

MEASUREMENTS OF PEERLESS AND ITALIAN FLUE AMERICAN DIRECT RADIATORS

Measurements are in Inches. See Outline, Figure 37.

Pattern and Catalog Height		A	C	E	F	G	Heating Surface, Sq. Ft.
Peerless, One-column,	38	37 13/32	31 3/32	4 13/32	5 3/8	2 1/2	3
	32	31 33/64	25 13/64	4 13/32	5 3/8	2 1/2	2 1/2
	26	25 25/32	19 15/32	4 13/32	5 3/8	2 1/2	2
	23	22 7/8	16 9/16	4 13/32	5 3/8	2 1/2	1 2/3
	20	19 61/64	13 41/64	4 13/32	5 3/8	2 1/2	1 1/2
Peerless, Two-column,	45	45 5/16	38 25/32	7 3/8	8 1/4	2 1/2	5
	38	37 5/8	31 3/32	7 3/8	8 1/4	2 1/2	4
	32	31 47/64	25 13/64	7 3/8	8 1/4	2 1/2	3 1/3
	26	26	19 15/32	7 3/8	8 1/4	2 1/2	2 2/3
	23	23 3/32	16 9/16	7 3/8	8 1/4	2 1/2	2 1/3
	20	20 11/64	13 41/64	7 3/8	8 1/4	2 1/2	2
	15	15	....	7 3/8	8 1/4	2 1/2	1 1/2
Peerless, Three-column,	45	45 3/4	38 25/32	9	9 3/4	2 1/2	6
	38	38 1/16	31 3/32	9	9 3/4	2 1/2	5
	32	32 11/64	25 13/64	9	9 3/4	2 1/2	4 1/2
	26	26 7/16	19 15/32	9	9 3/4	2 1/2	3 3/4
	22	22 3/16	15 7/32	9	9 3/4	2 1/2	3
	18	18 5/32	11 3/16	9	9 3/4	2 1/2	2 1/4
Peerless, Four-column,	45	46 1/64	38 25/32	10 1/2	11 1/4	3	10
	38	38 21/64	31 3/32	10 1/2	11 1/4	3	8
	32	32 7/16	25 13/64	10 1/2	11 1/4	3	6 1/2
	26	26 45/64	19 15/32	10 1/2	11 1/4	3	5
	22	21 29/64	15 7/32	10 1/2	11 1/4	3	4
	18	18 27/64	11 3/16	10 1/2	11 1/4	3	3
Italian Flue,	38	37 13/16	31 5/16	8 9/16	8 1/2	3	7
	32	31 29/32	25 7/16	8 9/16	8 1/2	3	5 3/4
	26	26	19 17/32	8 9/16	8 1/2	3	4 1/2
	20	20 1/16	13 17/32	8 9/16	8 1/2	3	3 1/4
Italian Flue, Ventilating,	39 1/2	39 3/8	31 5/16	8 9/16	8 1/2	3	7
	33 1/2	33 15/32	25 7/16	8 9/16	8 1/2	3	5 3/4
	27 1/2	27 9/16	19 17/32	8 9/16	8 1/2	3	4 1/2
	21 1/2	21 5/8	13 17/32	8 9/16	8 1/2	3	3 1/4
Ætna Flue, Window,	29	19 7/8	15 1/8	12 1/2	12 1/2	3	6
	18	17 15/16	13 1/8	12 1/2	12 1/2	3	5 1/3
	16	15 15/16	11 1/4	12 1/2	12 1/2	3	4 2/3
	14	13 15/16	9 3/16	12 1/2	12 1/2	3	4
	13	12 7/8	8 1/4	12 1/2	12 1/2	3	3 2/3

TABLE 24

DISTANCE FROM FLOOR TO CENTER OF LOWER TAPPINGS

Measurements are in Inches

Pattern	Water Supply and Return	Single-Pipe Steam	Two-Pipe Steam	
			Supply	Return
Peerless One-column	4 1/2	4	4 1/2	4
Peerless Two-column	4 1/2	4	4 1/2	4
Peerless Three-column	4 1/2	4	4 1/2	4
Peerless Four-column	4 1/2	4 1/2	4 1/2	4 1/2
Italian Flue	4 1/2	4	4 1/2	4
Italian Flue, Ventilating	6	5 1/2	6	5 1/2
Ætna Flue, Window	3	3	3	3

Wall Radiators. The *wall radiator* of cast iron (Fig. 42) has been developed in recent years until it is now nearly as efficient as the pipe-coil radiator, and much easier to handle because of the fact that it is made up in unit sections (Fig. 38) of approximately 5, 7, or 9 sq. ft. area, each

of which may be readily assembled by screw or push nipples to secure a single radiator of any desired amount of surface.

These radiators are assembled at the factory in stacks of from 3 to 5 sections for shipping, and these stacks may be connected later by right and left hexagon nut shoulder nipples, to obtain larger units, as shown in Figs. 39 and 40. Wall radiators are particularly useful for heating

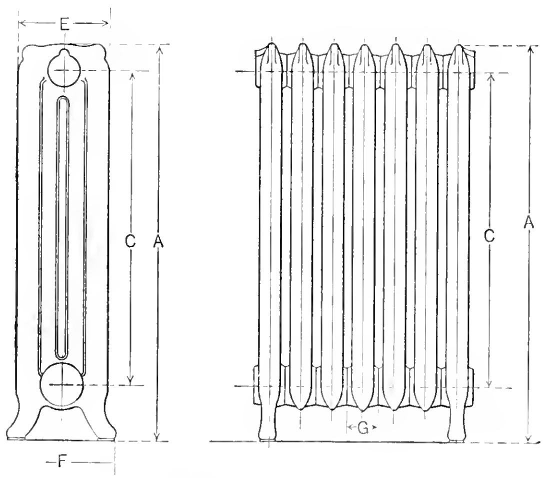


FIG. 37. MEASUREMENTS OF AMERICAN DIRECT RADIATORS.

A. Total height.
C. Distance from center of top to center of bottom opening of Water Radiators.
E. Width of sections.
F. Width at feet.
G. Distance from center to center of sections.

small rooms such as baths or toilets, and for use at the ceilings of basement rooms or just below skylights.

The method of *hanging* or *supporting wall radiators* is of great importance, as the tendency of these radiators to tear loose any brackets or hangers is very marked, due to the expansion and contraction strains produced by the piping system as well as by the radiators themselves. See " Radiator Accessories " in this chapter.

Wall sections are always assembled with bars *vertical* for greatest heating efficiency. Nos. 7-B and 9-B are regularly tapped as shown for connecting side by side. Nos. 5-A, 7-A, and 9-A are regularly tapped as shown for connecting end to end. No. 5-A can be furnished, when specially ordered, with tappings at 1, 2, 3, and 4. See Fig. 41 for additional data.

The manufacturer's data for the Rococo (Fig. 38) wall radiator are given in Fig. 41 and Table 25.

TABLE 25

AMERICAN ROCOCO WALL RADIATORS

Ratings and Measurements of Sections *American Radiator Co.*

Sections Number	Length, Inches	Width, Inches	Thickness, Inches	Thickness (with Bracket), Inches	Heating Surface, Sq. Ft.
5-A	16 5/8	13 5/16	2 7/8	3 1/2	5
7-A and 7-B	21 7/8	13 5/16	2 7/8	3 1/2	7
9-A and 9-B	29 1/16	13 5/16	2 7/8	3 1/2	9

Flue Radiators. The *flue radiator* of cast iron (Fig. 43) is used principally for direct-indirect heating, in which a large part of the heat from the radiator is given off by convection to heat either outside or recirculated air. This air is either supplied locally, through a suitable *inlet duct* from an *air inlet* in the outside wall or, if recirculated, it is taken from the room in which the radiator stands by reversing the *dampers* in the *box-base* under the radiator. See the chapter on "Gravity Indirect Heating by Steam and Water" for method of installation.

These radiators are assembled in the same manner as column radiators using screw or slip nipples. The data and dimensions for Italian Flue Radiators up to 10 sections are given in Table 26.

TABLE 26

ITALIAN FLUE VENTILATING RADIATORS†

For Steam and Water — *American Radiator Co.*

No. of Sections	*Length 3 Ins. per Section	Heating Surface—Square Feet			
		† 39½-In. Height	33½-In. Height	27½-In. Height	21½-In. Height
1	3	7 Sq. Ft. per Section	5¾ Sq. Ft. per Section	4½ Sq. Ft. per Section	3¼ Sq. Ft. per Section
2	6	14	11½	9	6½
3	9	21	17¼	13½	9¾
4	12	28	23	18	13
5	15	35	28¾	22½	16¼
6	18	42	34½	27	19½
7	21	49	40¼	31½	22¾
8	24	56	46	36	26
9	27	63	51¾	40½	29¼
10	30	70	57½	45	32½

NOTES

Tappings: 2 inches and bushed as specified.
Connections: Water—with extra heavy right and left threaded nipples at top and bottom. Steam—with extra heavy right and left threaded nipples at bottom only.
Low-Drip Hubs: One-Pipe Steam—on supply leg section. Two-Pipe Steam—on return leg section.
Measurements: Center of tappings to floors, and between centers of upper and lower water tappings, see Tables 23 and 24.

* Add ½ inch to length for each bushing.
† When the radiator is to be used *without* the box-bases, the leg sections are made 1½ inches *lower*, but the heating surface remains the same as given.

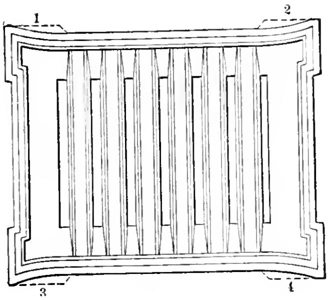
This pattern of radiator is not made in any special or odd shape, as the box-bases can only be made in straight form. It can be built up to any practical greater number of sections above 10.

Window Radiators. Low *window radiators* of cast iron are usually of the flue type (Fig. 44), and are designed to be placed below the stools or seats of very low windows. In construction they differ from the column radiators only in that the sections are usually flat hollow slabs with vertical ribs so cast upon them as to form flues of the interior surfaces of the radiator. This type is not made in heights above 20″, as shown by Table 27 giving data and dimensions.

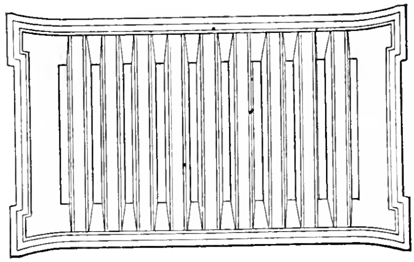
Special Radiators. Special radiators made up of the ordinary cast-iron sections have already been mentioned as possible shapes into which column radiators may be assembled in order to conform with special local conditions of installation.

The *circular radiator* is often placed around columns in entrance lobbies and can be obtained in the one-, two-, and three-column patterns only, in all regular heights, but usually only on special order and in strict conformity with the manufacturer's table of dimensions.

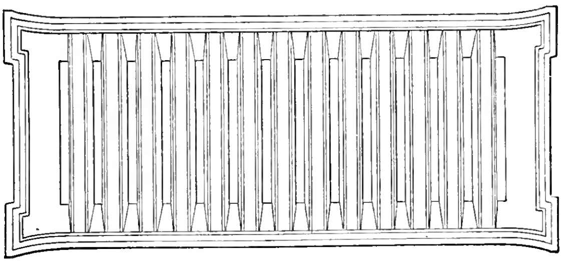
The *corner radiator* is also often used in entrance lobbies where the wall space is limited. It can be obtained in practically all patterns and heights except in the four-column and in the ventilating flue style. As from 3 to 5 sections are required to make the corner, depending on the style of radiator, it is necessary to specify the number of sections in each arm as well as in the



(5-*A*)



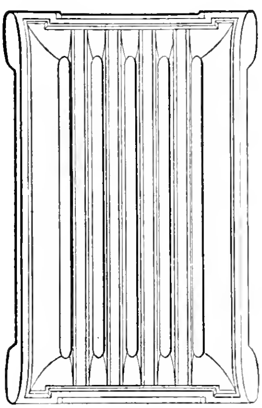
(7-*A*)



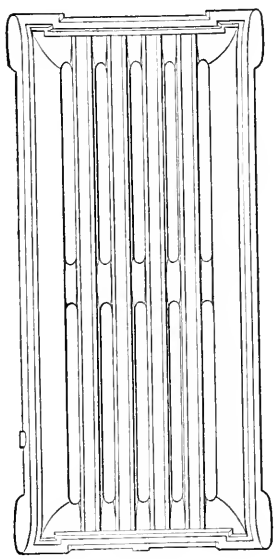
(9-*A*)

Rococo Wall Radiators

(*Screw Nipples.*)



(7-*B*)



(9-*B*)

Fig. 38. Rococo Wall Radiators.

corner. The supply and return ends must be indicated also to provide the proper hub for each end.

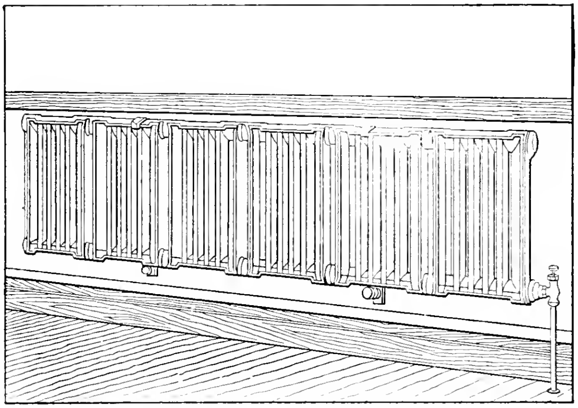


FIG. 42. TYPICAL INSTALLATION OF ROCOCO WALL RADIATORS IN SINGLE TIER ON ADJUSTABLE BRACKETS.

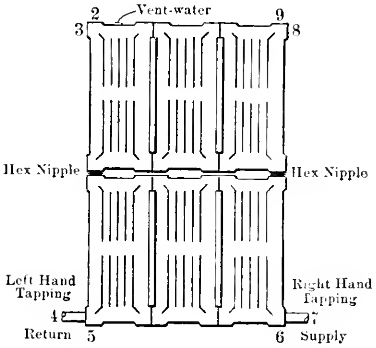


FIG. 39. WALL RADIATORS—SIX SECTIONS IN TWO TIERS—WATER.

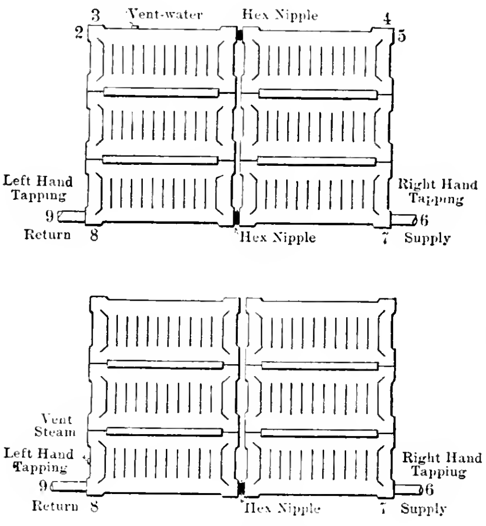


FIG. 40. WALL RADIATORS—SIX SECTIONS.

NOTE.—Water Radiator, above.
One or Two-Pipe Steam Radiator below.

The *hot-closet* or *dining-room radiator* is usually made up in the 38″ high 3-column pattern only, ranging from 53 to 113 sq. ft. of surface by increments of 10 sq. ft. The oven for the Rococo radiator is 14″ in width outside, and has two shelves 26½″ by 12½″ with 7″ between.

This oven fills the space between 38″ high end sections and short intermediate loop sections 22″ high.

Radiators without legs hung on concealed brackets may be obtained in the one-, two-, three-, and four-column patterns only, but of any regular heights (Fig. 45). Special brackets for each

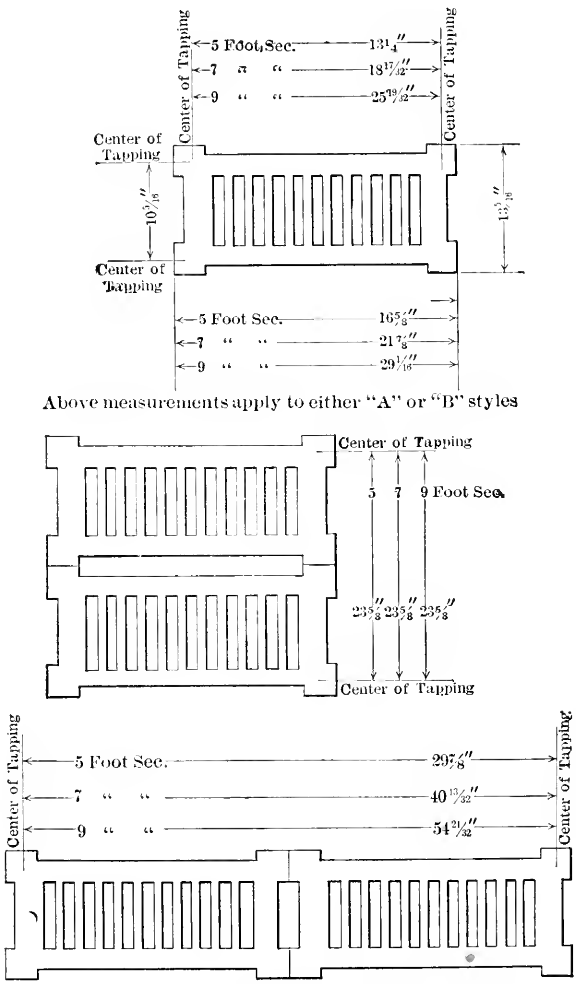


FIG. 41. ROCOCO WALL RADIATOR MEASUREMENTS.

width of radiator are required to support the radiator properly, and these must be firmly secured to the wall with the greatest care or else the radiator will pull them loose.

Other special radiators, such as *stairway* and *window radiators*, are made up of standard column sections of the same pattern and width, but of different heights, so selected as to conform to the irregular elevation of the space to be filled.

TABLE 27

ÆTNA FLUE, WINDOW RADIATORS

For Steam or Water *American Radiator Co.*

No. of Sections	*Length 3 Inches per Section	Heating Surface—Square Feet				
		20-In. Height	18-In. Height	16-In. Height	14-In. Height	13-In. Height
1	3	6 Sq. Ft. per Section	5⅓ Sq. Ft. per Section	4⅔ Sq. Ft. per Section	4 Sq. Ft. per Section	3⅔ Sq. Ft. per Section
2	6	12	10⅔	9⅓	8	7⅓
3	9	18	16	14	12	11
4	12	24	21⅓	18⅔	16	14⅔
5	15	30	26⅔	23⅓	20	18⅓
6	18	36	32	28	24	22
7	21	42	37⅓	32⅔	28	25⅔
8	24	48	42⅔	37⅓	32	29⅓
9	27	54	48	42	36	33
10	30	60	53⅓	46⅔	40	36⅔

NOTES

Can be built up to any practical greater number of sections above 10
Each section is 12½ inches wide.
Tappings: 2 inches and bushed as specified.
Connections: Extra heavy right and left threaded nipples at top and bottom.
Measurements: Center of tappings to floors, and between centers of upper and lower water tappings, see Tables 23 and 24.

* Add ½ inch to length for each bushing.

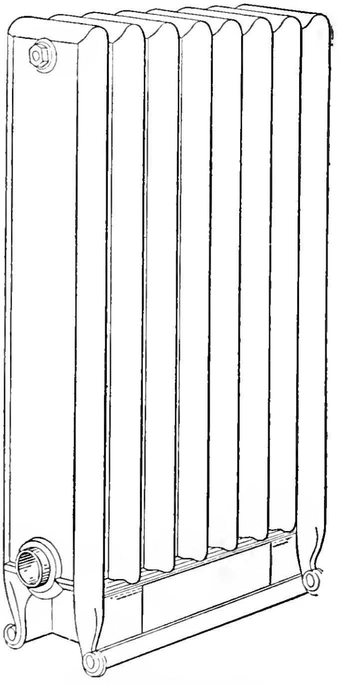


FIG. 43. ITALIAN FLUE, VENTILATING RADIATORS FOR STEAM AND WATER.

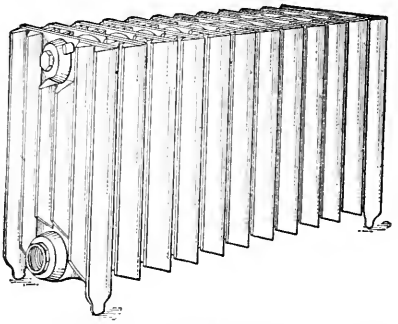


FIG. 44. ÆTNA FLUE, WINDOW RADIATOR FOR STEAM OR WATER. WIDTH 12½".

Pipe-Coil Radiators. Steel or wrought-iron direct radiators include practically all forms of direct radiators except those made of cast iron which have already been described.

These radiators are either made up out of steel or wrought-iron pipe with suitable headers or manifolds and known as pipe coils, or else are made of pressed steel or iron and known as pressed-metal radiators.

Pipe-coil radiators are in more or less general use in factory and industrial plants and are usually made up of 1″ or 1¼″ standard black pipe screwed into manifolds (Figs. 46 and 47). These coils may be of the *mitre* type as shown, or of the *return* or *box-coil* type using manifold tees or headers of cast iron in each case. A coil made up of *return bends* and straight pipe is sometimes used, and called a *trombone coil.* Its use is questionable since the long, continuous coil offers excessive friction to the steam or water. All of the above coils are intended to be hung on the side walls or from the ceiling, and suitable provision must be made for expansion in each case.

Vertical pipe radiators have also been used extensively in the past, and are made up by screwing short pieces of capped pipe into a cast-iron base forming a portable radiator very similar to the ordinary column radiator of cast iron.

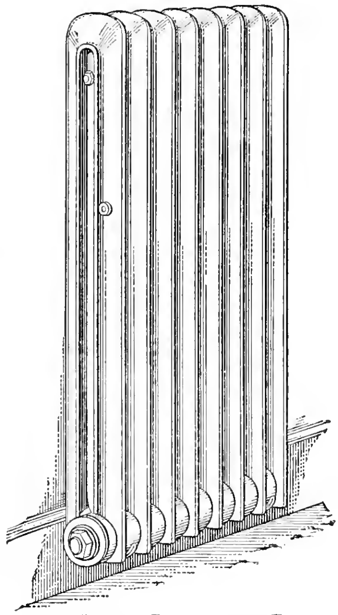


FIG. 45. DIRECT RADIATOR ON BRACKETS. (SINGLE COLUMN WITHOUT LEG SECTION.)

Pressed-metal Radiators. These radiators have been developed in recent years, and are most ingeniously fabricated of No. 20 U. S. standard gage soft iron (not steel) sheets made into shapes, widths, and heights which correspond almost exactly with the cast-iron column radiators.

Each section is made up of two pressed sheets joined by a double-lapped seam and the separate sections are connected by single-lapped seams. The pipe connection is made into a threaded malleable iron ring secured to the end section by rolling the sheet metal snugly over a suitable flange on the inner face of the ring. Air-valve connections are made in a similar manner. See Fig. 48 and Table 29.

These radiators are light in weight and therefore easy to handle and install, and cost less for freight and shipping charges. For the same height, width, and area of heating surface these radiators are shorter than cast-iron radiators, being spaced 1½″ instead of 2½″ center to center of sections. Because of their lightness these radiators are easily disturbed by jar or collision. The *U. S. Treasury Department* has used these radiators in certain classes of work, principally in hot-water heating.

TABLE 29

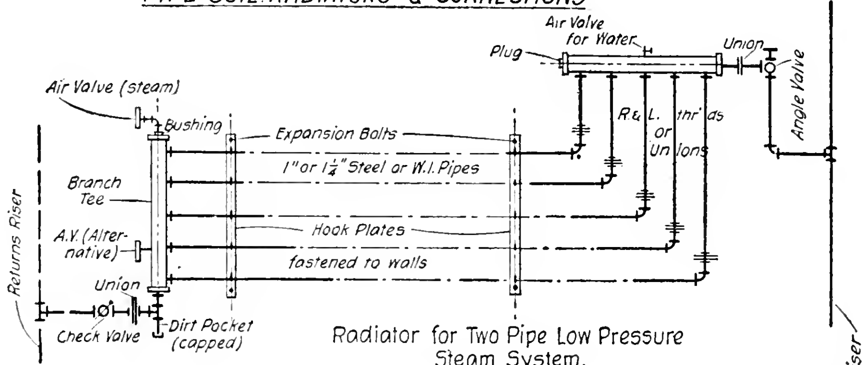
MEASUREMENTS OF PRESTO SINGLE-COLUMN RADIATORS

(See Fig. 48.) *Pressed Metal Radiator Co.*

A	B	C	D	E	F	G	H
Inches	Inches	Inches	Inches	Inches	Inches	Inches	Inches
32	29¾	25¾	1½	¾	4½	5½	4-5-6
26	23¾	19¾	1½	¾	4½	5½	4-5-6
23	20¾	16¾	1½	¾	4½	5½	4-5-6
20	17¾	13¾	1½	¾	4½	5½	4-5-6
17	14¾	10¾	1½	¾	4½	5½	4-5-6
14	11¾	7¾	1½	¾	4½	5½	4-5-6

Note—Allow ¾ inch for each bushing

PIPE COIL RADIATORS & CONNECTIONS



Radiator for Two Pipe Low Pressure Steam System.
Fig. 46

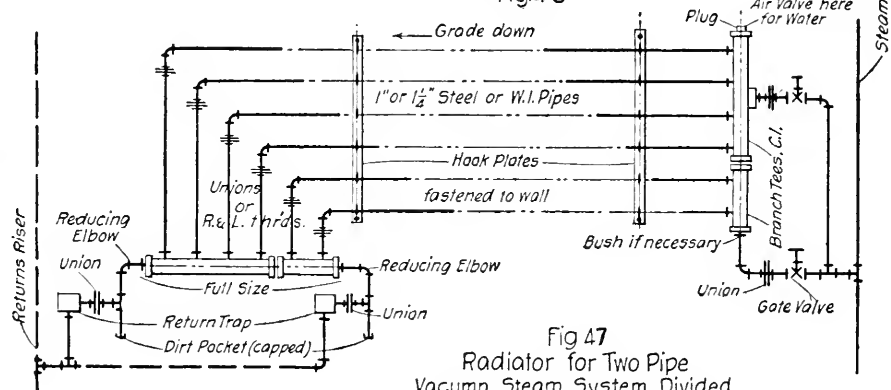
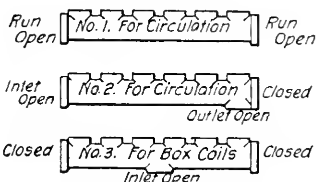


Fig 47
Radiator for Two Pipe Vacumn Steam System, Divided Surface

BRANCH TEES (Crane Co)



Order by size and number

Lineal feet of pipe per 1 ♯ H.S.

2.9' of 1" pipe = " "
2.3' " 1¼" " = " "
2.0' " 1½" " = " "
1.6' " 2" " = " "

TABLE 28

1" Branch Tees 2½" c-c			1¼" Branch Tees 3" c-c			1½" Branch Tees 3½" c-c			2" Branch Tees 4½" c-c		
Runs			Runs			Runs			Runs		
1"-1¼"	1½"	2"	1¼"-1½"	2"	2½"	1½"-2"	2½"	3"	2"	2½"-3"	3½"
No. of Branches			No. of Branches			No. of Branches			No. of Branches		
2 to 9	2 to 16	2 to 16	3 to 16	3 to 16	3 to 16	3 to 12	3 to 12	3 to 12	3 to 10	3 to 10	3 to 10
Inside Diams.			Inside Diams.			Inside Diams.			Inside Diams.		
1¾"	2¼"	2¼"	2½"	2½"	2½"	2¾"	2¾"	2¾"	3½"	3½"	3½"

Notes—All openings in Branch Tees for Circulation are tapped right hand.

Branch Tees for Box Coils are always tapped left hand in branches and right hand in back inlet.

The run and back opening of Branch Tees are tapped the same size as branches, unless otherwise ordered.

As already noted, the heat-transmission efficiency is practically the same as for cast-iron radiators of the same height and width.

The *dimensions and heating surface* of these radiators vary considerably from cast-iron

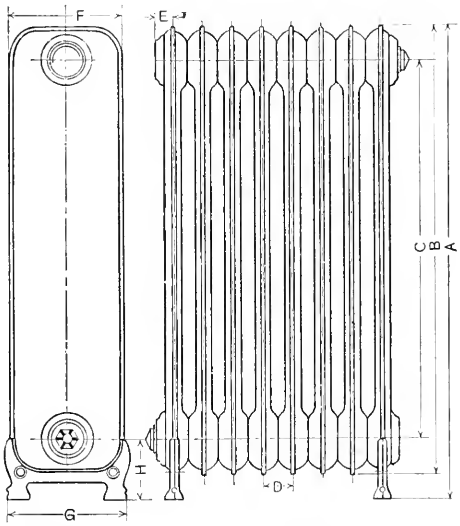


FIG. 48. PRESTO SINGLE COLUMN RADIATOR.

radiators of the same height and number of columns as indicated in Tables 29 and 30 for single-column, pressed-metal radiators.

For additional data and dimensions of the two-, three-, and four-column types see the catalog of the *Pressed Metal Radiator Co.*

TABLE 30

PRESTO—SINGLE-COLUMN FLOOR OR WALL RADIATORS FOR STEAM OR WATER

Each section is 4½ inches wide. Legs spread 5½ inches

No. of Sections	* Length 1½ Inch per Section	Heating Surface—Square Feet					
		32″ High	26″ High	23″ High	20″ High	17″ High	14″ High
		2 Sq. Ft. per Section	1.5 Sq. Ft. per Section	1.3 Sq. Ft. per Section	1.1 Sq. Ft. per Section	0.9 Sq. Ft. per Section	0.7 Sq. Ft. per Section
4	6	8	6.	5.2	4.4	3.6	2.8
5	7½	10	7.5	6.5	5.5	4.5	3.5
6	9	12	9.	7.8	6.6	5.4	4.2
7	10½	14	10.5	9.1	7.7	6.3	4.9
8	12	16	12.	10.4	8.8	7.2	5.6
9	13½	18	13.5	11.7	9.9	8.1	6.3
10	15	20	15.	13.	11.	9.	7.

* Length of radiator over all, including malleable iron hubs. Add ¾-inch for each bushing.
Legs are detachable and can be applied to any section.
Above radiators are tapped 1½-inches and bushed as specified.

Weights and Volumes of Direct Radiators. The weights and volumes of various types of direct radiators are often of considerable importance in heating problems, and these data are here given, as they are not ordinarily listed by the manufacturers. (Tables 31, 32, and 33.)

The *weights* given below indicate that direct cast-iron radiators may be estimated at about 7 lb. per sq. ft. of heating surface:

TABLE 31

WEIGHTS OF ROCOCO THREE-COLUMN STEAM AND WATER RADIATORS

Heights	Loop Section	Leg Section
nches	Pounds	Pounds
45	38	41
38	34	37
32	27	31
26	23	26
22	19	23
18	17	21

TABLE 32

WEIGHTS OF ITALIAN FLUE RADIATORS

Heights,	Loop Section		Leg Section	
	Steam	Water	Steam	Water
Inches	Pounds	Pounds	Pounds	Pounds
38	48	50	51	52
32	40	42	43	44
26	32	33	35	36
20	24	25	27	27

The weights of Rococo wall radiators full and empty, as well as the weight of the contained water, are as follows:

TABLE 33

WEIGHTS OF CAST-IRON ROCOCO WALL RADIATORS

	Area of Section		
	5 Sq. Ft.	7 Sq. Ft.	9 Sq. Ft.
	Pounds	Pounds	Pounds
Weight full of water	45.5	60.25	80.00
Weight empty	36.75	48.50	64.50
Weight of water	8.75	11.75	15.50
Weight per 1 square foot empty	7.35	6.93	7.17

The *internal volume of a radiator* is required for purposes of comparison and for computing the amount of water in a hot-water system. The following approximate values are generally used (Table 34):

TABLE 34

VOLUMES (INTERNAL) FOR VARIOUS RADIATORS

Kind of Radiator	Average Volume per Sq. Ft. of Surface
	Pints
Cast-iron column	1.5
Cast-iron flue	1.75
Cast-iron wall sections	1.65
Pipe, 1-inch diameter, steel or wrought iron	1.0
Pressed metal	1.0

The volume can only be accurately determined by filling a radiator of known heating surface with water and finding the weight of the water used.

Selection and Installation of Radiators. The selection of the proper type of radiation for actual conditions involves a great many factors and will depend very often upon the experience of the engineer who is designing the system.

First of all, the use of steam or water in the system must determine whether steam or water radiators are required, and if steam is used a low pressure gravity system will require steam radiators, while a vapor or vacuum system may permit either steam or water radiators to be used. Steam and water radiators are essentially different from each other, especially when made up in the cast-iron column type. Steam radiators of the one-, two-, and three-column pattern have nipples at the bottom of the section only, while all water radiators of whatever number of columns must have nipples at top and bottom of section to permit the proper collection and withdrawal of air from some one point at the top of the radiator. Fig. 36 shows such a radiator of four-column width with the water air valve tapping just above the plug in the end section. Radiators of four columns and above, however, have top and bottom nipples for steam as well as for water.

Steam radiators, therefore, *cannot be used for water* except in those patterns having both top and bottom nipples. Water radiators *are often used for steam* systems of the two-pipe vapor or vacuum type where the inlet is at the top on one end and the outlet at the bottom on the opposite end. This arrangement makes it very easy to operate the inlet valve if manually controlled. While water radiators can be used for low pressure gravity steam systems, they are not regarded as satisfactory because of air binding which is very liable to occur with one-pipe connections.

The *selection of column, wall, or flue radiators* of either cast iron or steel, or the *selection of pipe-coil radiators* for any given building must be determined largely by the character of the building and the use to which it is put.

Column radiators are generally used in residences, hotels, office buildings, hospitals, etc., while wall radiators are used largely in factory buildings, and in nearly all classes of work where the radiator must be hung on the side wall or ceiling. Wall radiators are often demanded on account of space limitations where otherwise column radiators would serve.

Flue radiators are used almost entirely for direct-indirect heating as already explained, although their use is justified wherever the radiator must give off the greater part of its heat by convection. Window radiators when placed under seats or otherwise concealed are often made of the flue type. Pipe-coil radiators are used for factory buildings, and indeed may be used for any work where cast-iron wall radiators would serve, provided sufficient space is available and their appearance is acceptable. Pipe-coil radiators usually require more wall area than cast-iron radiators for the same area of heating surface in the radiator.

The *selection of cast or pressed* metal radiators must depend on the experience of the designing engineer. The government departments usually specify cast-iron radiators.

The *availability, cost,* and *ease* of *installation* should also be considered in selecting a radiator. The standard radiator is the 38″ high 3-column type, and consequently it is generally in stock, and is usually the cheapest per sq. ft. of heating surface. Column radiators with feet are generally the easiest to install, as they arrive as a unit and merely have to be set in place on the floor.

Dividing the heating surface into two radiators to secure the benefit of manual control of the heat in mild weather is sometimes resorted to where a single radiator would otherwise be installed (Fig. 47). The customary division is to have one radiator of twice the heating surface of the other so that one, two, or three-thirds of the total surface may be turned on as required. Where thermostatic control is installed, this division is, of course, unnecessary. The operation of the above scheme depends entirely upon the interest of the occupant of the apartment, who must manipulate the valves.

The *use of mains and branches as radiating surface* is of very doubtful advantage or economy, and only in exceptional cases should they be left uncovered for this purpose. Heating by this means, if satisfactory in cold weather, will always result in overheating in mild weather since there is absolutely no way to regulate or control the amount of heat given off. It is far better to install an independent radiator and valve, and cover the mains and branches.

The *installation of radiators* requires both care and skill if satisfactory results and a workmanlike appearance are desired. The radiators must be delivered and handled with reasonable care.

Much trouble often develops in steam and water heating systems due to the careless dumping of radiators and boilers outside of, or in the buildings under construction. Frequently the castings are dumped off into a pile where it is an easy matter for particles of earth, sand, pebbles, lime, shavings, chips of wood, and other foreign materials to enter the tappings and later obstruct the system.

Long radiators of many sections must be handled with especial care during installation or broken or strained nipples will result and leaks will occur. For convenience in handling avoid

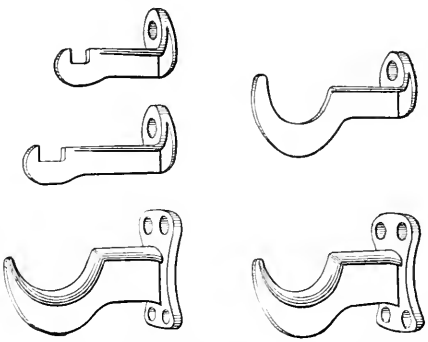


Fig. 49. Concealed Brackets for Steam and Water Column Radiators.

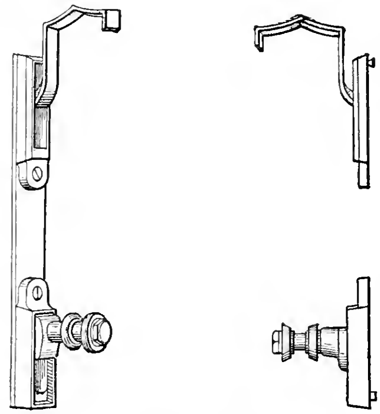


Fig. 50. Adjustable Wall Brackets for Wall Radiators.

ordering direct radiators in larger than a 32-section unit. Radiators of 1,000 pounds weight or over are liable to be strained or broken in transportation and it is recommended that these large radiators be shipped in halves.

Radiators should be placed from 2″ to 2½″ from the finished wall and made to center exactly with window, recess, or similar feature if placed in front of same.

The *finishing* or *bronzing* of radiators should be done only after the pressure and working tests have been made on the system and the heating units passed as satisfactory. Standard gold or aluminum bronze or paint should be applied after a priming coat free from oil has been given to the exposed surfaces.

Radiator Accessories. Radiator accessories including *brackets or hangers, shields, water-pans, foot-ups,* etc., are generally supplied by the radiator manufacturers, and must be adapted to each particular type of radiator.

The brackets or hangers for column radiators without feet or legs are usually made by each manufacturer to conform with his special line of radiators (Fig. 49). The brackets for wall radiators are quite different from those used for column radiators (Fig. 50).

It will be apparent that the brackets for the column radiators must be correctly spaced when put up, whereas the wall-radiator brackets are readily adjustable after installation and before the final pipe connections are made. The dimensions of brackets should be taken from the manufacturer's catalog in each case.

CHAPTER V

FUELS AND COMBUSTION

FUELS

Classification. Fuels are generally classified as solid, liquid, and gaseous.
Solid fuels are coal, wood, and wastes.
Liquid fuels are petroleum and its products.
Gaseous fuels are natural and artificial gas.

SOLID FUELS—COAL

The Formation of Coal. All coals are of vegetable origin, and are the remains of prehistoric forests. Destructive distillation, due to great pressures and temperatures, has resolved the organic matter into its invariable ultimate constituents, carbon, hydrogen, oxygen, and other substances, in varying proportions. The factors of time, depth of beds, disturbance of beds, and the intrusion of mineral matter resulting from such disturbances have produced the variation in the degree of evolution from vegetable fiber to hard coal. This variation is shown chiefly in the content of carbon, and Table 1 shows the steps of such variation.

The Composition of Coal. The uncombined carbon in coal is known as fixed carbon. Some of the carbon constituent is combined with hydrogen, and this, together with other gaseous substances driven off by the application of heat, forms that portion of the coal known as the volatile matter. The fixed carbon and the volatile matter constitute the combustible. The oxygen and nitrogen contained in the volatile matter are not combustible, but custom has applied this term to that portion of the coal which is dry and free from ash, thus including the oxygen and nitrogen in the combustible.

TABLE 1

APPROXIMATE CHEMICAL CHANGES FROM WOOD FIBER TO ANTHRACITE COAL

Substance	Carbon	Hydrogen	Oxygen
Wood Fiber	52.65	5.25	42.10
Peat	59.57	5.96	34.47
Lignite	66.04	5.27	28.69
Earthy Brown Coal	73.18	5.68	21.14
Bituminous Coal	75.06	5.84	19.10
Semi-Bituminous Coal	89.29	5.05	5.66
Anthracite Coal	91.58	3.96	4.46

Coals may be classified according to the percentages of fixed carbon and volatile matter contained in the combustible. *Wm. Kent* gives the following classification.

TABLE 2

CLASSIFICATION OF COALS

Name of Coal	Percentages of Combustible		B.t.u. per Pound of Combustible
	Fixed Carbon	Volatile Matter	
Anthracite	97.0 to 92.5	3.0 to 7.5	14,600 to 14,800
Semi-Anthracite	92.5 to 87.5	7.5 to 12.5	14,700 to 15,500
Semi-Bituminous	87.5 to 75.0	12.5 to 25.0	15,500 to 16,000
Bituminous, East	75.0 to 60.0	25.0 to 40.0	14,800 to 15,300
" West	65.0 to 50.0	35.0 to 50.0	13,500 to 14,800
Lignite	50.0 and under	50.0 and over	11,000 to 13,500

The non-combustible constituents are the ash and moisture, the former varying from 3 per cent to 30 per cent and the latter from 0.75 to 25 per cent of the total weight, depending on locality where mined and grade. A large percentage of ash is undesirable, as it not only reduces the calorific value of the fuel, but chokes up the air passages in the furnace and through the fuel bed, thus preventing the rapid combustion necessary to high efficiency. If the coal contains an excessive quantity of sulphur, trouble will result from its harmful action on the metal of the boiler where moisture is present, and because it unites with the ash to form a fusible slag or clinker which will choke up the great bars and form a solid mass in which large quantities of unconsumed carbon may be imbedded.

Moisture in coal may be more detrimental than ash in reducing the temperature of a furnace, as it is non-combustible, and absorbs heat both in being evaporated and superheated to the temperature of the furnace gases. In some instances, however, a certain amount of moisture in a bituminous coal produces a mechanical action that assists in the combustion and makes it possible to develop higher capacities than with dry coal.

General Characteristics of Hard and Soft Coals. The former contain fixed or uncombined carbon in large proportion, whereas the latter have an increasing percentage of carbon in combination with hydrogen, or hydrocarbon, which is volatile, and will distill off under high temperature, producing smoke. Hard coal usually contains more ash, especially in the smaller sizes. The distinguishing characteristics of the various coals are given in the following paragraphs as described in "Steam," *Babcock & Wilcox Co.*

Anthracite or Hard Coal. This coal ignites slowly, but when in a state of incandescence its radiant heat is very great. Its flame is very short and of a yellowish blue tinge and it can be burned with practically no smoke. This coal does not swell when burned although it contains from 3 to 7.5 per cent of volatile matter.

True or dry anthracite is characterized by few joints and clefts, and their squareness; great relative hardness and density; high specific gravity, ranging from 1.4 to 1.8, and a semi-metallic luster.

Anthracite is now classed and marketed according to graded sizes and designations as given in Table 3.

TABLE 3

NAMES AND SIZES OF ANTHRACITE OR "HARD" COAL

Names of Sizes	Will Pass Through	Will Not Pass Through
Buckwheat No. 1 / " No. 2	½-in. mesh	¼-in. mesh
or Rice	¼-in. mesh	⅛-in. mesh
Pea	¾-in. mesh	½-in. mesh
Chestnut, or Nut	1¼-in. mesh	¾-in. mesh
Stove or Range	1¾-in. mesh	1¼-in. mesh
Egg—in the East	2½-in. mesh	1¾-in. mesh
Large Egg—Chicago	4 -in. mesh	2¾-in. mesh
Small Egg—Chicago	2¾-in. mesh	2 -in. mesh
Broken, or Grate	4 -in. mesh	2½-in. mesh

The anthracite coals are, with some unimportant exceptions, confined to five small fields in eastern Pennsylvania.

Semi-Anthracite Coal. This coal kindles more readily, because of its higher content of volatile combustible, and burns more rapidly than anthracite. It has less density, hardness, and metallic luster than anthracite, and the average specific gravity is about 1.4.

This coal is found in the western part of the anthracite field in a few small areas.

Semi-Bituminous Coal. A softer coal than anthracite or semi-anthracite, contains more volatile hydrocarbon, and will kindle more easily and burn more rapidly. It is usually free burning, and, owing to its high calorific value, very desirable for steam-generation purposes.

This coal is found in Pennsylvania, Maryland, Virginia, West Virginia, and Tennessee.

Bituminous Coals. These coals are still softer than those described above and contain still more of the volatile hydrocarbons. The difference between the semi-bituminous and the bituminous coals is an important one, economically. The former have an average heating value per pound of combustible about 6 per cent higher than the latter, and they burn with much less smoke in ordinary furnaces. The distinctive characteristic of the bituminous coals is the omission of yellow flame and smoke when burning. In color they range from pitch black to dark brown, having a resinous luster in the most compact specimens, and a silky luster in such specimens as show traces of vegetable fiber. The specific gravity is ordinarily about 1.3.

Bituminous coals are either of the *caking* or *non-caking* variety. The former, when heated, fuse and swell in size; the latter burn freely, do not fuse, and are commonly known as *free burning* coals. Caking coals are rich in volatile hydrocarbons, and are valuable in gas manufacture.

Bituminous coals *absorb moisture* from the atmosphere. The surface moisture can be removed by ordinary drying, but a portion of the water can be removed only by heating the coal to a temperature of about 250° F.

TABLE 4

NAMES AND SIZES OF BITUMINOUS OR "SOFT" COAL

For "Domestic" soft coals there are no uniform names and sizes, but they are marketed in the various states under about these classes:

"Screenings" usually smallest sizes.
"Duff" goes through ⅛-inch screen.
"No. 3 Nut" goes through 1¼-in. screen, over ¾-inch screen.
"No. 2 Nut" goes through 2-inch screen, over 1¼-inch screen.
"No. 1 Domestic Nut" goes through 3-inch screen, over 1½- or 2-inch screen.
"No. 4 Washed" goes through ¾-inch screen, over ¼-inch screen.
"No. 3 Washed Chestnut" goes through 1¼-inch screen, over ¾-inch screen.
"No. 2 Washed Stove" goes through 2-inch screen, over 1¼-inch screen.
"No. 1 Washed Egg" goes through 3-inch screen, over 2-inch screen.
"No. 3 Roller Screened Nut" goes through 1½-inch screen, over 1-inch screen.
"No. 2 Roller Screened Nut" goes through 2-inch screen, over 1½-inch screen.
"No. 1 Roller Screened Nut" goes through 3½-inch screen, over 2-inch screen.
"Egg" goes through 6-inch, over 3-inch screen.
"Lump" or "Block" goes through 6-inch screen, or over.
"Run-of-Mine" in fine and large lumps.
Pocahontas Smokeless: generally sized as: "Nut," "Egg," "Lump," and "Mine-Run."

Bituminous coal is far more generally distributed than any of the other coals, being found in the Appalachian field in the states of Pennsylvania, West Virginia, Maryland, Virginia, Ohio, Kentucky, Tennessee, and Alabama; a field nearly 900 miles in length. The eastern interior field includes Michigan, all of Illinois, and parts of Indiana and Kentucky. The western field includes Iowa, Missouri, Kansas, Oklahoma, Arkansas, and Texas. The Rocky Mountain fields include parts of Montana, Wyoming, Colorado, Utah, and New Mexico. The Pacific Coast fields are limited to small areas in California, Oregon, and Washington.

Cannel Coal. This is a variety of bituminous coal, rich in hydrogen and hydrocarbons and is exceedingly valuable as a gas coal. It has a dull, resinous luster and burns with a bright flame without fusing. Cannel coal is seldom used for steam coal, though it is sometimes mixed with semi-bituminous coal where an increased economy at high rates of combustion is desired. The composition of cannel coal is approximately as follows: fixed carbon, 26 to 55 per cent; volatile matter, 42 to 64 per cent; earthy matter, 2 to 14 per cent. Its specific gravity is approximately 1.24.

Names and Sizes of Cannel Coal: For fireplace, "Hand-Picked Lump"; for stoves, "Egg."

Lignite. Organic matter in the earlier stages of its conversion into coal is known as lignite and includes all varieties which are intermediate between peat and coal of the older formation. Its specific gravity is low, being 1.2 to 1.23, and when freshly mined it may contain as high as 50 per cent of moisture. Its appearance varies from a light brown, showing a distinctly woody structure, in the poorer varieties, to a black, with a pitchy luster resembling hard coal, in the best varieties. It is non-caking and burns with a bright but slightly smoky flame with moderate heat. It is easily broken, will not stand much handling in transportation, and if exposed to the weather will rapidly disintegrate, which will increase the difficulty of burning it.

Its composition varies over wide limits. The ash may run as low as 1 per cent and as high as 50 per cent. Its high content of moisture and the large quantity of air necessary for its combustion cause large stack losses. It is distinctly *a low-grade fuel,* and is used almost entirely in the districts where mined, because of its cheapness.

Lignites resemble the brown coals of Europe and are found in the western states of Wyoming, New Mexico, Arizona, Utah, Montana, North Dakota, Nevada, California, Oregon, and Washington. Many of the fields given as those containing bituminous coals in the western states also contain true lignite. Lignite is also found in the eastern part of Texas and in Oklahoma.

Peat. This is organic matter in the first stages of its conversion into coal and is found in bogs and similar places. Its *moisture* content when cut is extremely high, averaging 75 to 80 per cent. It is unsuitable for fuel until dried, and even then will contain as much as 30 per cent moisture. Its ash content when dry varies from 3 to 12 per cent. In this country, though large deposits of peat have been found, it has not as yet been found practicable to utilize it for steam-generating purposes in competition with coal. In some European countries, however, the peat industry is common.

Pressed Fuels. In this class are those fuels composed of the dust of some suitable combustible, pressed and cemented together by a substance possessing binding, and in most cases, inflammable properties. Such fuels, known as *briquettes,* are extensively used in foreign countries and consist of carbon or soft coal, too small to be burned in the ordinary way, mixed usually with pitch or coal tar. Much experimenting has been done in this country in briquetting fuels, the government having taken an active interest in the question, but as yet this class of fuel has not come into common use, as the cost and difficulty of manufacture and handling have made it impossible to place it in the market at a price to compete successfully with coal.

Coke. This is a porous product, consisting almost entirely of carbon, remaining after certain manufacturing processes have distilled off the hydrocarbon gases of the fuel used. It is produced (1) from gas coal distilled in gas retorts; (2) from gas or ordinary bituminous coals burned in special furnaces called coke ovens; and (3) from petroleum by carrying the distillation of the residuum to a red heat.

Coke is a *smokeless* fuel. It readily *absorbs moisture* from the atmosphere and if not kept under cover its moisture content may be as much as 20 per cent of its own weight.

Gas-house coke is generally softer and more porous than oven coke, ignites more readily, and requires less draft for its combustion.

Names and Sizes of Domestic By-Product Coke: "Egg," 3-in. to 2½-in. "Large Stove," 2½-in. to 2-in. "Small Stove," 2-in. to 1½-in. "Nut," 1½-in. to ¾-in. "Pea," ¾-in. to ½-in.

The *heat values* of coke range from 12,500 B.t.u. per 1 lb. to 13,500 B.t.u., depending on the ash content, which may vary from 5 to 10 per cent.

Coal Analysis. The *analysis of a coal* should be ascertained if possible. The actual composition of any coal is determined by an ultimate chemical analysis, which can only be made by an experienced chemist.

The *ultimate analysis* of a fuel gives the percentage by weight of the various elements composing same. Such an analysis is usually reported on the dry sample as 100 per cent, and the percentage of moisture in the original sample given separately.

The true analysis is easily obtained by dividing each reported percentage by 100 plus the percentage of H_2O in the original sample as indicated in Table 5.

The *proximate analysis* of a fuel gives the percentage by weight of the fixed carbon, volatile matter, moisture, and ash.

The *moisture* is found by heating a finely pulverized sample (through a 100-mesh sieve) for one hour in a drying oven at a temperature of 240° to 280° F. The loss in weight in this time is due to moisture.

The sample is then heated to a red heat for several hours in a closed crucible to expel the *volatile matter* (gases). Weighings are made at intervals and no air is allowed to come in contact with sample until a constant minimum weight is reached.

TABLE 5

TYPICAL ULTIMATE ANALYSIS

Constituent	Chemist's Report (Based on Dry Fuel)	True Analysis (Fuel as Received)
Carbon	76.91%	72.52%
Hydrogen	5.07	4.78
Oxygen	8.65	8.156
Nitrogen	1.16	1.09
Sulphur	1.21	1.14
Ash	7.00	6.60
	100.00	
Moisture	6.06	5.714
	106.06%	100.00%

Finally the sample is heated to a white heat and the *fixed carbon* allowed to combine with the oxygen of the air, forming carbon dioxide gas (CO_2).

The residue remaining is *ash* or *incombustible*, and if a careful record of weighings has been

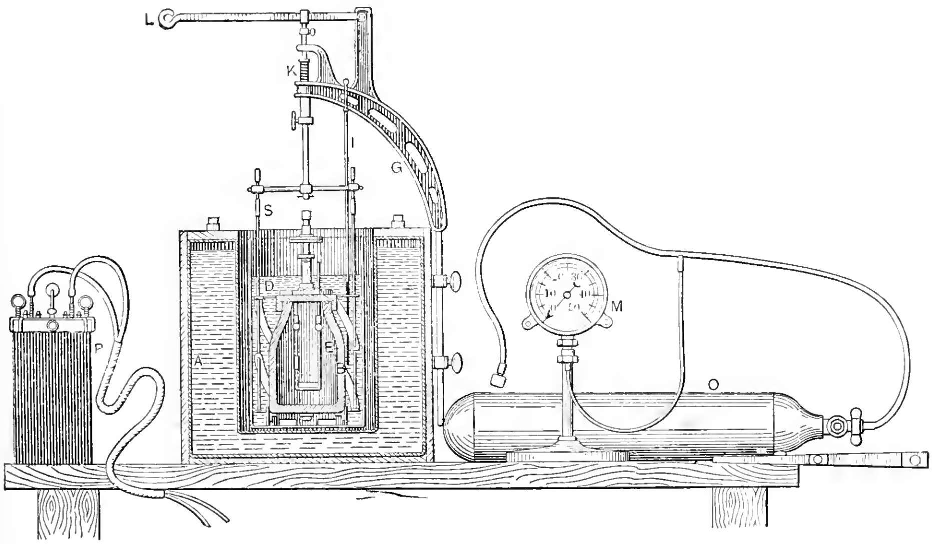


FIG. 1. MAHLER-BOMB CALORIMETER.

made the loss of weight for each step represents successively moisture, volatile matter, fixed carbon, and the final residue, the ash.

See Table 6 for results of proximate analyses on Anthracite and Semi-Anthracites.

Heat Value of a Fuel. The *heat of combustion or calorific value of a fuel* is the number of B.t.u. evolved when 1 lb. of the fuel is completely burned in air or oxygen.

A fuel *calorimeter* is used to determine the heat generated by the combustion of a known weight of the fuel, and this heat reduced to a pound basis. In the case of a solid or liquid fuel a *bomb calorimeter* (Fig. 1) is employed, and the standard apparatus in use at the present time is essentially the same as that devised by *M. Pierre Mahler*.

In such an apparatus the fuel is completely burned, and the heat generated by the combustion is absorbed by water, the amount of heat being calculated from the increase in the temperature of the water. A calorimeter which has been accepted as the best for such work is one in which the fuel is burned in a steel bomb filled with compressed oxygen. The function of the oxygen, which is ordinarily under a pressure of about 25 atmospheres, is to cause the rapid and complete combustion of the fuel sample. The fuel is ignited by means of an electric current, allowance being made for the heat produced by such current, and by the burning of the fuse wire.

The apparatus consists of: A water jacket, *A*, which maintains constant conditions outside of the calorimeter proper, and thus makes possible a more accurate computation of radiation losses.

The porcelain-lined steel bomb, *B*, in which the combustion of the fuel takes place in compressed oxygen.

The platinum pan, *C*, for holding the fuel.

The calorimeter proper, *D*, surrounding the bomb and containing a definite weighed amount of water.

An electrode, *E*, connecting with the fuse wire, *F*, for igniting the fuel placed in the pan, *C*.

A support, *G*, for a water agitator.

A thermometer, *I*, for temperature determination of the water in the calorimeter. The thermometer is best supported by a stand independent of the calorimeter, so that it may not be moved by tremors in the parts of the calorimeter, which would render the making of readings difficult. To insure accuracy, readings should be made through a telescope or eyeglass.

A spring and screw device for revolving the agitator.

A lever, *L*, by the movement of which the agitator is revolved.

A pressure gage, *M*, for noting the pressure of the oxygen admitted to the bomb. Between 20 and 25 atmospheres are ordinarily employed.

An oxygen tank, *O*.

TABLE 6

COMPOSITION AND HEAT VALUES OF ANTHRACITE COALS

Locality	Fixed Carbon	Volatile	Moisture	Ash	Sulphur	B.t.u. per Lb. of Dry Coal
Anthracite						
Pennsylvania	78.60	.....	....	14.80	0.40	.....
Buckwheat	81.32	3.84	3.88	10.96	0.67	12,200
Wilkesbarre	76.94	6.42	1.34	15.30	....	11,801
Scranton	79.23	3.73	3.33	13.70	....	12,149
Scranton	84.46	5.37	0.97	9.20	....	12,294
Cross Creek	89.19	1.96	3.62	5.23	....	13,723
Lehigh Valley	75.20	7.36	1.44	16.00	....	12,423
Lykens Valley	76.94	6.21	....	.....	....	15,300
Lykens Valley	81.00	5.00	....	.....	....	15,300
Wharton	86.40	3.08	3.71	6.22	0.58	15,000
Buck Mt.	82.66	3.95	3.04	9.88	0.46	15,070
Beaver Meadow	88.94	2.38	1.50	7.11	0.01	.....
Lackawanna	87.74	3.91	2.12	6.35	0.12	.....
Rhode Island	85.00	.....	....	7.00	0.90	.....
Arkansas	74.49	14.73	1.52	9.26	....	13,217
Semi-Anthracite						
Pennsylvania, Loyalsock	83.34	8.10	1.30	6.23	1.03	15,400
Bernice	82.52	3.56	0.96	3.27	0.24	15,050
Bernice	89.39	8.56	0.97	9.34	1.04	15,475
Wilkesbarre	88.90	7.68	....	3.49	....	14,199
Lycoming Creek	71.53	13.84	0.67	13.96	0.03	.....
Virginia, Natural Coke	75.08	12.44	1.12	11.38	0.47	.....
Arkansas	74.06	14.93	1.35	9.66	....	.....
Indian Territory	73.21	13.65	5.11	8.03	1.18	13,662
Maryland, Easby	83.60	16.40	....	.....	....	11,207

A battery or batteries, *P*, the current from which heats the fuse wire used to ignite the fuel.

This or a similar calorimeter may be used in the determination of the heat of combustion of solid or liquid fuels. Whatever the fuel to be tested, too much importance cannot be given to

the securing of an average sample. Where coal is to be tested, tests should be made from a portion of the dried and pulverized laboratory sample, the methods of obtaining which have been described. In considering the methods of calorimeter determination, the remarks applied to coal are equally applicable to any solid fuel, and such changes in methods as are necessary for liquid fuels will be self-evident from the same description.

A considerably simpler form of apparatus has been perfected by *Professor S. W. Parr*, which depends upon the oxidizing effect of sodium peroxide to "burn" the fuel. The results are not as accurate as those obtained with the *Mahler* apparatus, but serve for many classes of commercial work.

Heat values of typical American coals are given in Tables 6 and 7 as determined by the Mahler-bomb calorimeter.

TABLE 7

HEAT VALUES OF BITUMINOUS COALS

From selected free-burning and caking soft fuels taken from *U. S. Geological Survey* Bulletin No. 332, and *U. S. Bureau of Mines* Bulletin No. 23

State	Test No.	Kind of Fuel	County	B.t.u. per Lb. Dry Coal
Alabama	375	Soft—caking	Bibb	13,671
Alabama	484	Soft—free burning	Jefferson	14,447
Arkansas	293	Soft—caking	Sebastian	13,705
Arkansas	308	Semi-anthracite—caking	Johnson	14,125
Arkansas	340	Lignite	Quachita	9,549
Georgia	481	Soft—free burning	Chattooga	12,865
Illinois	448	Soft—free burning	Williamson	12,920
Illinois	511	Soft briquettes	St. Clair	13,271
Illinois	509	Soft—caking	Saline	13,621
Indiana	428	Soft—free burning	Greene	13,099
Indiana	435	Soft—caking	Pike	13,545
Indiana	464	Soft briquettes	Parke	11,930
Indian Territory	437	Soft—free burning	....	13,932
Indian Territory	449	Semi-anthracite	....	14,682
Kansas	311	Soft—free burning	Linn	12,343
Kentucky	434	Soft—free burning	Union	14,026
Maryland	490	Soft—free burning	Allegany	14,515
Maryland	518	Soft briquettes	Allegany	14,717
Missouri	319	Soft—caking	Randolph	11,747
Montana	477	Lignite—free burning	Carbon	11,628
New Mexico	392	Soft—caking	Colfax	13,059
New Mexico	387	Soft—free burning	Colfax	12,721
Ohio	483	Soft—free burning	Belmont	13,381
Pennsylvania	473	Soft—caking	Indiana	14,240
Pennsylvania	499	Soft—free burning	Cambria	14,119
Pennsylvania	514	Soft briquettes	Westmoreland	14,382
Tennessee	409	Soft briquettes	Claiborne	14,092
Tennessee	368	Soft—free burning	Campbell	14,008
Tennessee	363	Soft—caking	Grundy	13,257
Texas	291	Lignite—free burning	Wood	11,131
Utah	404	Soft—free burning	Summit	12,586
Virginia	482	Anthracite—free burning	Montgomery	12,679
Virginia	507	Soft—caking	Tazewell	14,177
Washington	290	Subbit—free burning	King	11,772
Washington	359	Soft—free burning	Kittitas	12,996
West Virginia	305	Soft—free burning	Marion	13,964
West Virginia	439	Soft—caking	Kanawha	13,995
Wyoming	399	Soft—free burning	Carbon	12,222
Wyoming	400	Subbit—free burning	Unita	12,488

NOTE.—The above values were obtained at the *St. Louis Testing Plant* from 139 samples of coal. The heating values of the various coals were established by "actually burning one gram of the air-dried coal in oxygen in a Mahler-bomb calorimeter." These values in B.t.u. give the theoretical maximum thermal value of soft coals.

High and Low Heat Value of Fuels. For any fuel containing hydrogen the calorific value as found by the calorimeter is higher than can be realized under most working conditions existing in boiler practice by an amount equal to the latent heat of the water formed by combustion. This heat would reappear if the vapor was condensed, but in ordinary practice the vapor passes away uncondensed. This fact gives rise to a distinction in heat values between the so-called "higher" and "lower" calorific values. The higher value, *i.e.*, the one determined by the

calorimeter, is the proper scientific unit, is the value which should be used in boiler testing work, and is the one recommended by the *American Society of Mechanical Engineers*.

TABLE 8

HEAT VALUES OF ILLINOIS COALS

Field* Designation	Geological Seam Number	Name of Field	Moisture, Percent	Ash in Dry Coal, Percent	B.T.U. PER POUND	
					Moist Coal	Dry Coal
A	1	Rock Island	11.57	6.27	11,915	13,473
B	2	Wilmington	15.34	5.87	11,316	13,367
B	2	Northern	14.86	10.08	11,054	12,983
F	5	Springfield	12.66	12.31	10,990	12,583
D	5	Peoria and Fulton	14.67	15.10	10,381	12,166
N	5	Saline	5.90	8.98	12,552	13,197
E	6	Grape Creek	12.76	8.59	11,500	13,181
G	6	Virden	14.38	11.69	10,774	12,584
H	6	Pana	14.38	11.69	10,774	12,584
I	6	Central Illinois	14.38	11.69	10,774	12,584
J	6	Centralia	14.38	11.69	10,774	12,584
M	6	Big Muddy	14.38	11.69	10,774	12,584
K	6	Du Quoin	10.36	13.86	10,967	12,235
L	7	Williamson and Franklin	9.65	12.16	11,508	12,737

For screenings (slack), increase values of ash about 20 per cent, and decrease heating values about 5 per cent.
* See Fig. 2 (map from "Data").

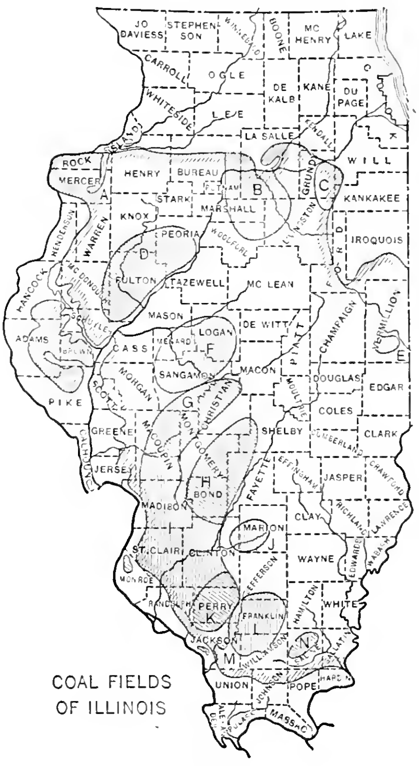


FIG. 2.

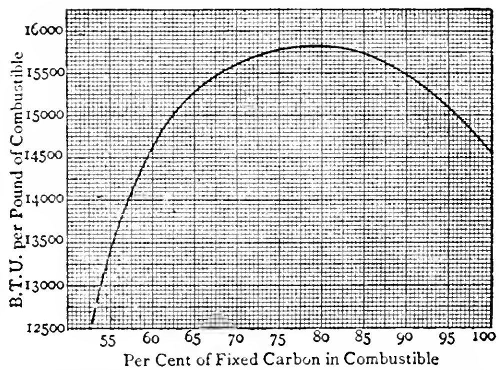


FIG. 3. GRAPHIC REPRESENTATION OF RELATION BETWEEN HEAT VALUE PER POUND OF COMBUSTIBLE AND FIXED CARBON IN COMBUSTIBLE AS DEDUCED BY WM. KENT.

There is no absolute measure of the lower heat value, and in view of the wide difference in opinion among physicists as to the deductions to be made from the higher or absolute unit in this determination, the lower value must be considered an artificial unit. The lower value entails the use of an ultimate analysis and involves assumptions that would make the employment of such a unit impracticable for commercial work. The use of the low value may also lead to error and is not recommended for boiler practice.

An example of its illogical use may be shown

by the consideration of a boiler operated in connection with a special economizer where the vapor produced by hydrogen is partially condensed by the economizer. If the low value were used in computing the boiler efficiency, it is obvious that the total efficiency of the combined boiler and economizer must be in error through crediting the combination with the heat imparted in condensing the vapor and not charging such heat to the heat value of the coal.

Calorific Value by Formula. The following expression known as *Du Long's* formula for heating value per pound of coal can be used if the ultimate analysis of the fuel is known:

$$F = 14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\ S,$$

where C, H, O, and S represent the proportionate parts of each element per 1 lb. of fuel, and F denotes the heat value in B.t.u. per pound due to combustion.

This formula does not apply when the fuel contains carbon monoxide, CO, but can be made to apply by adding a term, 10,150 C, in which C is the proportionate part of carbon burned to monoxide.

Example. Application of formula to a coal of ultimate analysis as here given follows:

Analysis (Based on fuel as received)

C	74.79%
H	4.98
O	6.42
N	1.20
S	3.24
H_2O	1.55
Ash	7.82
	100.00%

Then by *Du Long's* formula

$$14{,}600 \times 0.7479 + 62{,}000\left(0.0498 - \frac{0.0642}{8}\right) + 4{,}000 \times 0.0324 = 13{,}650 \text{ B.t.u. per 1 lb. coal.}$$

A bomb-calorimeter test showed 13,480 B.t.u. for this coal. The formula fails to allow for evaporating and superheating the moisture present in the fuel.

Heat Value Based on Fixed Carbon. The relation between the heat value per pound of combustible and the fixed carbon in the combustible is shown by Fig. 3 as deduced by *Wm. Kent.*

Calorific Value of Gaseous Fuels. The *calculation of the calorific value of gaseous fuels* may be made by means of *Du Long's* formula provided the constituent gases are separated into their elementary gases and a term is added to provide for carbon monoxide, or the calculation may be based on the percentages of the constituent gases present and the heat value of each, as given in the following table:

TABLE 9

WEIGHT AND CALORIFIC VALUE OF VARIOUS GASES AT 32° F. AND ATMOSPHERIC PRESSURE WITH THEORETICAL AMOUNT OF AIR REQUIRED FOR COMBUSTION

Gas	Symbol	Cubic Feet of Gas per Pound	B.t.u. per Pound	B.t.u. per Cubic Feet	Cubic Feet of Air Required per Pound of Gas	Cubic Feet of Air Required per Cubic Foot of Gas
Hydrogen	H	177.90	62000	349	428.25	2.41
Carbon monoxide	CO	12.81	4450	347	30.60	2.39
Methane	CH_4	22.37	23550	1053	214.00	9.57
Acetylene	C_2H_2	13.79	21465	1556	164.87	11.93
Olefiant gas	C_2H_4	12.80	21440	1675	183.60	14.33
Ethane	C_2H_6	11.94	22230	1862	199.88	16.74

Example. Assume a natural gas, the analysis of which in percentages by volume is oxygen = 0.40, carbon monoxide = 0.95, carbon dioxide = 0.34, olefiant gas (C_2H_4) = 0.66, ethane (C_2H_6) = 3.55, marsh gas (CH_4) = 72.15, and hydrogen = 21.95. All but the oxygen and the carbon dioxide are combustibles, and the heat value per cubic foot will be:

$$
\begin{array}{lrcl}
\text{From} & CO & = 0.0095 \times 347 = & 3.22 \\
 & C_2H_4 & = 0.0066 \times 1675 = & 11.05 \\
 & C_2H_6 & = 0.0355 \times 1862 = & 65.99 \\
 & CH_4 & = 0.7215 \times 1053 = & 757.58 \\
 & H & = 0.2195 \times 349 = & \underline{75.95} \\
\text{B.t.u. per cu. ft.} & & = & 913.79
\end{array}
$$

The net air required for combustion of one cubic foot of the gas will be:

$$
\begin{array}{lrl}
 & CO & = 0.0095 \times 2.39 = 0.02 \\
 & C_2H_4 & = 0.0066 \times 14.33 = 0.09 \\
 & C_2H_6 & = 0.0355 \times 16.72 = 0.59 \\
 & CH_4 & = 0.7215 \times 9.54 = 6.88 \\
 & H & = 0.2195 \times 2.39 = \underline{0.52} \\
\text{Total net air per cu. ft.} & & = 8.10
\end{array}
$$

LIQUID FUELS—OIL

Petroleum. The following distinguishing characteristics of petroleum have been taken from "Steam," *Babcock & Wilcox Co.:*

"Petroleum is practically the only liquid fuel sufficiently abundant and cheap to be used for the generation of steam. It possesses many advantages over coal and is extensively used in many localities.

"There are three kinds of petroleum in use, namely, those yielding on distillation: 1st, paraffin; 2nd, asphalt; 3rd, olefine. To the first group belong the oils of the Appalachian Range and the Middle West of the United States. These are a dark brown in color with a greenish tinge. Upon their distillation such a variety of valuable light oils are obtained that their use as fuel is prohibitive because of price.

"To the second group belong the oils found in Texas and California. These vary in color from a reddish brown to a jet black and are used very largely as fuel.

"The third group comprises the oils from Russia, which, like the second, are used largely for fuel purposes.

"The light and easily ignited constituents of petroleum, such as naphtha, gasoline, and kerosene, are oftentimes driven off by a partial distillation, these products being of greater value for other purposes than for use as fuel. This partial distillation does not decrease the value of petroleum as a fuel; in fact, the residuum known in trade as *fuel oil* has a slightly higher calorific value than petroleum and because of its higher flash point, it may be more safely handled. Statements made with reference to petroleum apply as well to fuel oil.

"In general, crude oil consists of carbon and hydrogen, though it also contains varying quantities of moisture, sulphur, nitrogen, arsenic, phosphorus, and silt. The moisture contained may vary from less than 1 to over 30 per cent, depending upon the care taken to separate the water from the oil in pumping from the well. As in any fuel, this moisture affects the available heat of the oil, and in contracting for the purchase of fuel of this nature it is well to limit the percentage of moisture it may contain. A large portion of any contained moisture can be separated by settling and for this reason sufficient storage capacity should be supplied to provide time for such action."

The *calorific values of petroleum* range from 18,000 to 22,000 B.t.u. per pound, and the percentage composition and other data are given in Table 10. The *flash point* of crude oil is the temperature at which it begins to give off inflammable gases. This temperature varies greatly for different oils, as shown in the table.

The *fire point* is the temperature at which these gases are liberated in sufficient quantity to burn continuously.

TABLE 10

COMPOSITION AND CALORIFIC VALUE OF VARIOUS OILS

Kind of Oil	Per Cent Carbon	Per Cent Hydrogen	Per Cent Sulphur	Per Cent Oxygen	Specific Gravity	Deg. Flash Point	B.t.u. per Pound	Authority
California†	85.04	11.52	2.45	0.99*	.....	...	17871	B. & W. Co.
California	81.52	11.51	0.55	6.92*	.....	230	18667	U. S. N.‡
Texas	87.15	12.33	0.32	....	0.908	370	19338	U. S. N.
Texas	87.29	12.32	0.43	....	0.910	375	19659	U. S. N.
Ohio	83.4	14.7	0.6	1.3	.....	...	19580	....
Pennsylvania	84.9	13.7	....	1.4	0.886	...	19210	Booth
West Virginia	84.3	14.1	....	1.6	0.841	...	21240	....
Mexico	....	.....	....	....	0.921	162	18840	B. & W. Co.
Russia	....	.....	....	....	.....	...	.....	....
Caucasus	86.6	12.3	....	1.10	0.938	...	20138	....
Java	87.1	12.0	....	0.9	0.923	...	21163	....
Austria	....	.....	....	....	.....	...	.....	....
Galicia	82.2	12.1	5.7	....	0.870	...	18416	....
Italy, Parma	84.0	13.4	1.8	....	0.786	...	.....	....
Borneo	85.7	11.0	....	3.31	.....	...	19240	Orde

* Includes N. † Per cent moisture = 1.40. ‡ Liquid Fuel Board.

The *comparative value of petroleum and coal as fuel* may be summed up to the advantage of the liquid fuel as follows: The cost of handling is much lower, both in delivery and in burning same, while for equal heat value much less storage space is required, and this space may be at a distance from the boilers. Higher efficiencies are obtainable, since the combustion is more

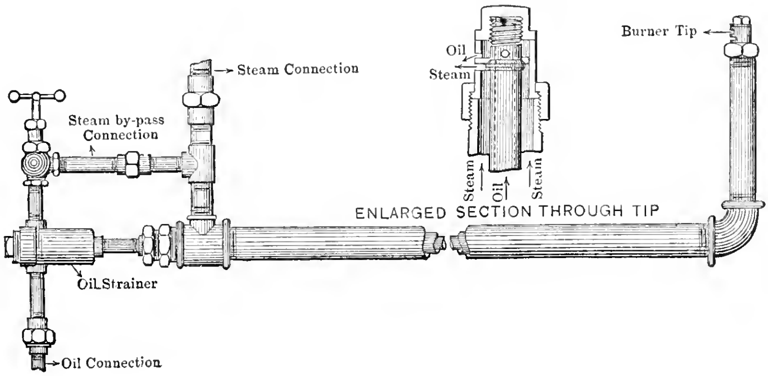


FIG. 4. PEABODY OIL BURNER.

perfect, less excess air is required, temperatures are more constant, and since smoke is largely eliminated, the heating surfaces are correspondingly clean.

The intensity of the fire can be instantly regulated to suit the load requirements, and there is no deterioration from loss of heat value by disintegration due to storage.

The *disadvantage of the liquid fuel* arises from the fact that the oil must have a reasonably high flash point to reduce the danger of explosion, and city ordinances may, in certain cases make its use practically prohibitive. Owing to the high temperatures of the oil flame the boiler upkeep cost may be increased.

The comparative evaporative power of coal and oil is given in Table 11

TABLE 11

EVAPORATION OF WATER FROM COAL AND OIL

Taken from the "U. S. Geological Report on Petroleum" for 1900

Designation of Coal NOTE.—One ton coal = 2,000 lb. One barrel oil = 42 gals. or 336 lb. One gallon oil = 8 lb.	Pounds of Water Evaporated from and at 212° per Pound of Combustible in the Coal	Barrels of Petroleum Required to do same Amount of Evaporation as 1 Ton of Coal Petroleum 18° to 40° Baumé
Pittsburg lump and nut, Pennsylvania	10.0	4.0
Pittsburg nut and slack, Pennsylvania	8.0	3.2
Anthracite, Pennsylvania	9.8	3.9
Indiana Block	9.5	3.8
Georges Creek lump, Maryland	10.0	4.0
New River, West Virginia	9.7	3.8
Pocahontas lump, West Virginia	10.5	4.2
Cardiff lump, Wales	10.0	4.0
Cape Breton, Canada	9.2	3.7
Nanaimo, British Columbia	7.3	2.9
Co-operative, British Columbia	8.9	3.6
Greta, Washington	7.6	3.0
Carbon Hill, Washington	7.6	3.0

Under favorable conditions 1 pound of oil will evaporate from 14 to 16 pounds of water from and at 212°; 1 pound of coal will evaporate from 7 to 10 pounds of water from and at 212°; 1 pound of natural gas will evaporate from 18 to 20 pounds of water from and at 212°.

Oil Burning. The burning of petroleum fuel or oil can only be accomplished in steam-boiler practice by the use of suitable burners, which must atomize the oil so thoroughly that each particle will be brought into contact with the minimum quantity of air necessary for its complete combustion before the gases come in contact with any heating surfaces. The furnace must be of highly refractory material, the radiant heat from which will assist in the combustion. No localization of the heat must occur at the heating surfaces or trouble will result from overheating and blistering.

The *burners* may be classified under three general types: 1st, *spray burners,* in which the oil is atomized by steam or compressed air; 2nd, *vapor burners,* in which the oil is converted into vapor and then passed into the furnace; 3rd, *mechanical burners,* in which the oil is atomized by submitting it to high pressure and passing it through a small orifice.

The *Peabody Burner* (Fig. 4) is of the latter type. These mechanical burners have been in general use only a short time in this country, and the round-flame burner has proved more satisfactory than the flat-flame burner of this type.

The *efficiency of oil burning* with boilers of 500 horsepower may run as high as 83 per cent gross or 81 per cent net after deducting 2 per cent for steam used by burner. The conditions of average practice are such that efficiencies ranging from 5 to 10 per cent less than the above are about the best that may be expected.

GASEOUS FUELS

The gaseous fuels in most common use are blast furnace gas, natural gas, and by-product coke-oven gas.

Blast Furnace Gas. This is a by-product from the blast furnace of the iron industry; the composition of a typical sample from a *Bessemer Furnace* is as follows:

$$CO_2 = 10.0\%,\ CO = 26.2,\ H = 3.1,\ CH_4 = 0.2,\ N = 60.5.$$

With the exception of the small amount of carbon in combination with hydrogen as methane, and a very small percentage of free hydrogen, ordinarily less than 0.1 per cent, the calorific value

of blast furnace gas is due to the CO content which when united with sufficient oxygen as used under a boiler, finally burns to CO_2. The heat value of such gas will vary in most cases from 85 to 100 B.t.u. per cubic foot under standard conditions. In modern practice, where the blast is heated by hot blast stoves, approximately 15 per cent of the total amount of gas is used for this purpose, leaving 85 per cent of the total for use under the boilers or in gas engines, that is, approximately 8500 pounds of gas per ton of pig iron produced. In a modern blast furnace plant, the gas serves ordinarily as the only fuel required.

Natural Gas. This gas has a limited use but is, of course, confined to restricted areas. The best results are secured by using a large number of small burners to which the gas is supplied at a pressure of about 8 ounces. The calculations for amount of gas required to give a certain heating effect should in all cases be based on volume reduced to standard conditions of temperature and pressure, namely, 32°F., and 14.7 lb. pressure per sq. in.

The variation in composition and heating value of natural gas is shown in the following table:

TABLE 12

TYPICAL ANALYSIS (BY VOLUME) AND CALORIFIC VALUES OF NATURAL GAS FROM VARIOUS LOCALITIES

Locality of Well	H	CH_4	CO	CO_2	N	O	Heavy-Hydro-Carbons	H_2S	B.t.u. per Cu. Ft. Calculated*
Anderson, Ind.	1.86	93.07	0.73	0.26	3.02	0.42	0.47	0.15	1017
Findlay, O.	1.64	93.35	0.41	0.25	3.41	0.39	0.35	0.20	1011
St. Ive, Pa.	6.10	75.54	Trace	0.34	.....	....	18.12	....	1117
Pittsburgh, Pa.	9.64	57.85	1.00	....	23.41	2.10	6.00	....	718
Pittsburgh, Pa.	20.02	72.18	1.00	0.80	.....	1.10	4.30	....	917

* B.t.u. calculated, using percentages of constituent gases, and separate heat values.

By-product Coke-Oven Gas. This is also known as *artificial gas, or illuminating gas*, and is a product of the destructive distillation of coal in a distilling or by-product coke oven. In this class of apparatus the gases, instead of being burned at the point of their origin, as in a beehive or retort coke oven, are taken from the oven through an uptake pipe, cooled, and yield as by-products: tar, ammonia, and illuminating and fuel gas. A certain portion of the gas product is burned in the ovens and the remainder used or sold for illuminating or fuel purposes, the methods of utilizing the gas varying with plant operation and locality.

Table 13 gives the analyses and heat value of certain samples of by-product coke-oven gas utilized for fuel purposes.

This gas is nearer to natural gas in its heat value than is blast furnace gas, and, in general, the remarks as to the proper methods of burning natural gas and the features to be followed in furnace design hold as well for by-product coke-oven gas.

TABLE 13

TYPICAL ANALYSIS OF BY-PRODUCT COKE-OVEN GAS

Sample No.	CO_2	O	CO	CH_4	H	N	B.t.u. per Cu. Ft.
1	0.75	Trace	6.0	28.15	53.0	12.1	505
2	2.00	Trace	3.2	18.80	57.2	18.0	399
3	3.20	0.4	6.3	29.60	41.6	16.1	551
4	0.80	1.6	4.9	28.40	54.2	10.1	460

The essential difference in burning the two fuels is the pressure under which it reaches the gas burner. Where this is ordinarily from 4 to 8 ounces in the case of natural gas, it is approxi-

mately 4 inches of water in the case of by-product coke-oven gas. This necessitates the use of larger gas openings in the burners for the latter class of fuel than for the former.

By-product coke-oven gas comes to the burners saturated with moisture, and provision should be made for the blowing out of water of condensation. This gas, too, carries a large proportion of tar and hydrocarbons which form a deposit in the burners, and provision should be made for cleaning this out. This is best accomplished by an attachment which permits the blowing out of the burners by steam.

FUEL CONSUMPTION

Methods of Estimation. The *estimated fuel consumption for heating boilers* per heating season may be based on grate areas, sq. ft. of radiation installed, or cubic contents of building to be heated. The *U. S. Treasury Department* allows 5 tons of coal per sq. ft. of grate area per season of 240 days, or 1 lb. of coal per cu. ft. of contents of building for the same period. This applies to government buildings. The district steam-heating companies estimate 500 lb. of steam per sq. ft. of direct steam radiation per season, which is practically the same as 70 lb. of coal of good quality.

This is approximately equivalent to assuming that one-third of the radiation installed is in operation continuously for 240 days, and reference to Fig. 5 will show that this assumption is justified in view of the average outside winter temperatures existing in any locality.

In this chart the normal or average heating requirements as shown by the shaded area are 31.9 per cent of the maximum demand for the entire season as represented by the rectangle marked by the heavy dash lines.

In other words, the coal required for a heating season is about one-third the quantity that would be used if all the radiation were in constant use every hour of the day and night.

The *U. S. Weather Bureau* publishes data available to any one on application, and from these it is possible to get an idea of the average or normal temperature which has prevailed in different sections of the country, and knowing the conditions under which the heating installation has been guaranteed, a proportion may be established for average conditions.

Fuel Consumption Based on Average Temperature. In arriving at a proportion it should be understood that the temperature of air surrounding the cooling or condensing surface is a constant (say 70°) and that the temperature of the water or steam leaving the boiler is a constant (180° for water, 220° for steam), under which conditions heat loss from cooling or condensing surface will be a constant (say 150 B.t.u. for water, 250 B.t.u. for steam). For example: An installation is made with the specific purpose of heating to 70° with an outside temperature of 20° below zero—with 2 lb. pressure at boiler or 220°. What are the heating service requirements under average conditions with outside temperature 40° above zero and room temperature 70°? The heat loss from the radiator being the same the load will vary directly as the difference between the outside and inside temperatures

or, $\dfrac{\text{max. load}}{\text{avg. load}} = \dfrac{70 - (-20)}{70 - (40)} = \dfrac{90}{30} = 3$, that is, the average load will be under above conditions, one-third the maximum load; and for practical approximation the amount of coal used for average conditions will be one-third of that used under maximum conditions.

The amount of coal for maximum conditions is determined as follows:

Since each foot of direct steam radiation or its equivalent will give off 250 B.t.u. per hour under conditions of 2 pounds (220°) pressure at boiler, 70° air surrounding the direct radiators (the piping on the average job may be roughly taken as 25 per cent of the direct radiation), and, since for approximation we may assume 8000 B.t.u. per pound of anthracite coal burned, we can readily estimate the amount of coal per hour if R = amount of direct radiation in square feet. (1) $\dfrac{1.25 \times R \times 250}{8000} = C$ = coal per hour in pounds. In a heating season of 7 months or 210 days of 24 hours each, there would be burned under maximum conditions during the en-

tire period, (2) $\frac{1.25 \times R \times 250 \times 210 \times 24}{8000 \times 2000}$ = maximum tons of coal. But we have seen above that if the installation is figured for 20° below zero and the average temperature is 40°, only ⅓ the maximum is burned. Then for average conditions we would have (3) $\frac{1.25\ R \times 250 \times 210 \times 24}{8000 \times 2000 \times 3}$ = tons burned per season approximately. If $R = 1$, then this equation resolves itself into .0328 tons per foot of radiation per season or 3.28 tons per hundred feet ($R = 100$).

For water-heating under similar conditions (except that temperature of heating medium

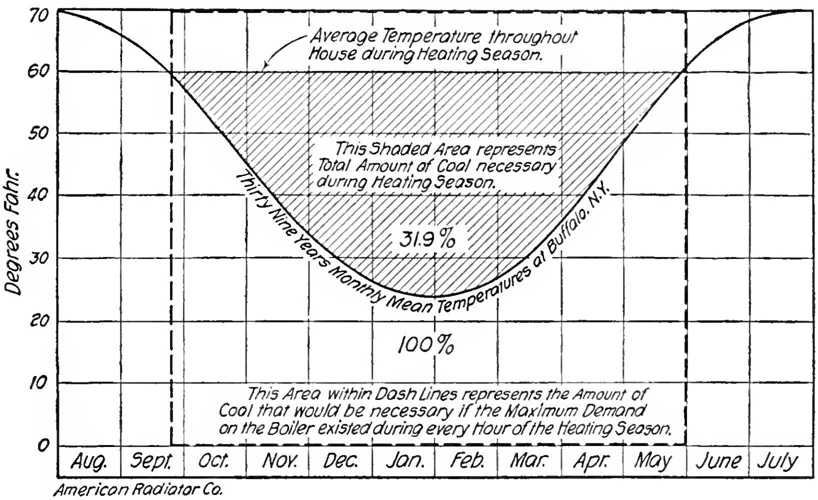


FIG. 5.

would be 180° at boiler and loss per square foot of radiation 150 B.t.u. instead of 250 B.t.u.), the coal burned would be as $\frac{150}{250}$ or 60 per cent of 3.28 = 1.91 tons per hundred feet of radiation per season.

By the above method an approximation may be arrived at for any month if the average or normal temperature for that month be obtained from the Weather Bureau records.

In the same way the amount of soft coal used may be approximated by using 5800 instead of 8000 B.t.u. per pound, and where *down-draft boilers* are used with free-burning soft coal as fuel 7200 B.t.u. may be used instead of 8000.

Example. Approximately how much fuel—free-burning soft coal—would be required in an installation containing 3,000 square feet of direct steam radiation exclusive of piping? Radiation figured to be ample to maintain a temperature of 70° inside with weather outside 20° below zero—steam at boiler 220°—plant located in Chicago, where the average or mean temperature during the seven heating months is 36° (Weather Bureau records 33 years). Boiler of down-draft type having proper conditions of installation and attention.

$$\frac{3000 \times 1.25 \times 250 \times 24 \times 210 \times (70 - 36)}{7200 \times 2000\ (70 - (-20))} = 124 \text{ tons per season or } 4.13 \text{ tons per 100 feet of radiation.}$$

It will of course be appreciated that the above method should not be taken as giving exact

figures as to how much coal should be used but can only be considered an approximation based on average instead of exact conditions.

Fuel Storage. Space for fuel storage must be based on fuel consumption per season as estimated above, and in government buildings it is customary to proportion the storage space on the basis of 8 sq. ft. of floor area per ton, the storage space being made ample to hold an entire season's supply.

The following volumes per ton of 2240 lb. of coal are given for proportioning storage space.

Bituminous Coal, 41 to 45 cu. ft., and may run as high as 49 cu. ft.

Anthracite Coal, 34 to 41 cu. ft.

Charcoal, 123 cu. ft.

Coke, 70.9 cu. ft.

The above is based on fuel broken down ready for market. Also 1 bushel hard coal = 86 lb. and 1 bushel soft coal = 76 lb.

Fuel Selection. The fuel selected for a given heating plant is usually the one which can be obtained at the lowest price for a given heating effect, although in residence and small apartment house plants considerations of cleanliness, and uniform combustion over long periods may demand that a more expensive fuel be burned. In general, however, the fuel which can be bought for *the fewest cents per 1,000 B.t.u. transmitted* to the steam and water in the type of boiler which is installed should be selected.

The *U. S. Treasury Dept.* observes the following practice in selecting fuels, grates, and boilers for heating plants. Anthracite coal is given the preference, where costs are equal or nearly so, and in small plants it is always used even at a higher cost, especially where the firing must be done at considerable intervals.

Soft coal is used, however, in all localities remote from the anthracite regions, and is burned on a plain grate in plants having a boiler capacity of less than 2600 sq. ft. of steam radiation. For boilers of larger capacity than this, using soft coal, a down-draft furnace is installed, or else a smokeless type of firebox boiler with a down-draft water-grate is selected.

Small size anthracite requires that the maximum openings in grate bars shall not exceed $\frac{3}{16}$ in. in width, and it has been found necessary to add 40 per cent to the actual direct radiation installed, to get the rating in selecting the proper size of boiler when burning this fuel.

With bituminous coal 35 per cent is added to the actual direct radiation installed to get the rating in selecting the proper size of boiler required.

Oil is used as fuel when 3 times the cost of one barrel plus the cost of 7 kw.-hrs. (usually 70 cents) of electric current per day is less than the cost of one ton of coal. In this case the boiler is equipped with oil burners, pump, tanks, etc., and its size is determined by adding 25 per cent to the actual direct radiation installed, in order to get the rating.

Gas is used as fuel when the cost of 20,000 cu. ft. of natural gas is cheaper than one ton of coal, and the boiler is equipped with burners, pilot light and governor. Size of boiler is determined by adding 25 per cent to the actual direct radiation installed to get the rating.

The Comparative Cost of Heating by Coal, Gas, or Electricity. This cost will depend on the relative costs of these agents in any given locality, and a selection may be arrived at or the actual cost determined as shown below, using prices applying to the locality.

Heating by Electricity. The heating value of one kilowatt-hour is approximately 3,415 thermal units—therefore, at 10c per kw.-hr., one cent will purchase 341.5 thermal units.

At $7.50 per ton hard coal, having available about 8,000 thermal units per pound, one cent will purchase 21,333 B.t.u. At this rate it would cost sixty-two and seven-tenths times as much to heat with electricity as with coal.

Heating by Gas. The available heating value of one cubic foot of gas for heating purposes is approximately 600 B.t.u. per cubic foot. At 50c per 1,000 cubic feet, one cent would purchase 12,000 thermal units.

With coal at $7.50 per ton, as above, it would cost one and eight-tenths times as much to heat with gas as with hard coal.

With electricity and gas on the same basis, as above, but with soft coal, having a heating value of 6,000 thermal units per pound and selling at $3.50 per ton, it would cost one hundred times as much to heat with electricity as with soft coal, and two and nine-tenths times as much to heat with gas as with soft coal.

COMBUSTION

Combustion of Fuel. Combustion as used in steam engineering signifies a rapid chemical combination between oxygen, and the carbon, hydrogen and sulphur composing the various fuels. This combination takes place usually at high temperature with the evolution of light and heat.

The substance combining with the oxygen is known as the *combustible*, and if it is completely burned or oxidized the combustion is *perfect*, that is, no more oxygen can be taken up by the products of the reaction.

The combustion is *imperfect* or incomplete when carbon burns to form carbon monoxide, CO, instead of the dioxide, CO_2, since the former may be further burned to form carbon dioxide if the necessary oxygen is supplied.

The temperature at which the reaction begins to take place is known as the *kindling temperature* and is different for each combustible. The following values are from *Stromeyer:*

TABLE 14

KINDLING TEMPERATURES

Fuel	Temp. F.	Fuel	Temp. F.
Lignite dust	300° F.	Coke	Red Heat
Dried peat	435	Anthracite	Red Heat 750°F.
Sulphur	470	Carbon monoxide	Red Heat 1211
Anthracite dust	570	Hydrogen	1030–1290
Coal	600		

Combustion takes place only between hot gases and oxygen, hence all combustibles are practically gaseous at the instant of combustion.

The characteristics of these gases and atmospheric air must be definitely known before combustion problems can be solved, and such data will be found in the following tables:

TABLE 15

DENSITY OF GASES AT 32° F. AND ATMOSPHERIC PRESSURE 29.92 INS.
(ADAPTED FROM SMITHSONIAN TABLES)

Gas	Chemical Symbol	Specific Gravity, Air = 1	Weight of One Cubic Foot, Pounds	Volume of One Pound, Cubic Feet	Relative Density, Hydrogen = 1	
					Exact	Appr.
Oxygen	O	1.053	0.08922	11.208	15.87	16
Nitrogen	N	0.9673	.07829	12.773	13.92	14
Hydrogen	H	0.0696	.005621	177.90	1.00	1
Carbon dioxide	CO_2	1.5291	.12269	8.151	21.83	22
Carbon monoxide	CO	0.9672	.07807	12.809	13.89	14
Methane	CH_4	0.5576	.04470	22.371	7.95	8
Ethane	C_2H_6	1.075	.08379	11.935	14.91	15
Acetylene	C_2H_2	0.920	.07254	13.785	12.91	13
Sulphur dioxide	SO_2	2.2639	.17862	5.598	31.96	32
Air	....	1.0000	.08071	12.390	....	....

Combustion Reactions. The constituent elements of a gas combine with oxygen in perfectly definite proportions by weight and volume, forming definite *combustion products*. These reactions as well as the proportions in which the gases combine have been tabulated for use in computation work and are given herewith.

TABLE 16

OXYGEN AND AIR REQUIRED FOR COMBUSTION

BY WEIGHT

At 32° F. and 29.92 Inches

1	2	3	4	5	6	7	8	9	10
Oxidizable Substance or Combustible	Chemical Symbol	Atomic or Combining Wgt.	Chemical Reaction	Product of Combustion	Oxygen per Pound of Column 1 in Pounds	Nitrogen per Pound of Column 1 = 3.32*×O in Pounds	Air per Pound of Column 1 = 4.32†×O in Pounds	Gaseous Product per Pound of Col. 1 = 1+ Col. 8 in Lbs.	Heat Value per Pound of Col. 1 in B.t.u.
Carbon....	C	12	$C+2O=CO_2$	Carbon dioxide...	2.667	8.85	11.52	12.52	14600
Carbon....	C	12	$C+O=CO$	Carbon monoxide.	1.333	4.43	5.76	6.76	4450
Carbon monoxide	CO	28	$CO+O=CO_2$	Carbon dioxide...	0.571	1.90	2.47	3.47	10150‡
Hydrogen..	H	1	$2H+O=H_2O$	Water...........	8.0	26.56	34.56	35.56	62000
Methane...	CH_4	16	$CH_4+4O=CO_2+2H_2O$	Carbon dioxide and water.....	4.0	13.28	17.28	18.28	23550
Sulphur...	S	32	$S+2O=SO_2$	Sulphur dioxide ..	1.0	3.32	4.32	5.32	4050

* Ratio by weight of N to O in air.
† 4.32 pounds of air contain one pound of O.
‡ Per pound of C in the CO.

TABLE 16 (*Continued*)

BY VOLUME

1	2	11	12	13	14	15	16	17	18
Oxidizable Substance or Combustible	Chemical Symbol	Volumes of Column 1 Entering Combination Volume	Volumes of Oxygen Combining with Column 11 Volume	Volumes Product Formed Volume	Volume per Lb. of Column 1 in Gaseous Form, Cu. Ft.	Volume of Oxygen per Pound of Column 1, Cu. Ft.	Volume of Products of Combustion per Pound of Column 1, Cu. Ft.	Volume of Nitrogen per Pound of Column 1 = 3.782*× Column 15, Cu. Ft.	Volume of Gas per Pound of Column 1 = Column 16 + Column 17, Cu. Ft.
Carbon........	C	1	2	$2CO_2$	14.95	29.89	29.89	112.98	142.87
Carbon........	C	1	1	2CO	14.95	14.95	29.89	56.49	86.38
Carbon monoxide	CO	2	1	$2CO_2$	12.80	6.40	12.80	24.20	37.00
Hydrogen......	H	2	1	$2H_2O$	179.32	89.66	179.32	339.09	518.41
Methane.......	CH_4	1	2	$1CO_2+2H_2O$	22.41	44.83	67.34	169.55	236.89
Sulphur........	S	1	2	$2SO_2$	5.60	11.21	11.21	42.39	53.60

* Ratio by volume of N to O in air. *Babcock & Wilcox Co.*

It will be seen from this table that a pound of carbon will unite with 2⅔ pounds of oxygen to form carbon dioxide, and will evolve 14,600 B.t.u. As an intermediate step, a pound of carbon may unite with 1⅓ pounds of oxygen to form carbon monoxide and evolve 4450 B.t.u., but in its further conversion to CO_2 it would unite with an additional 1⅓ times its weight of oxygen and evolve the remaining 10,150 B.t.u.

When a pound of CO burns to CO_2, however, only 4350 B.t.u. are evolved, since the pound of CO contains but $^3/_7$ lb. carbon.

Air Required for Combustion. It has already been shown that each combustible element in the fuel will unite with a definite amount of oxygen. With the ultimate analysis of the fuel known, the theoretical amount of air required for combustion may be readily calculated.

Example. Let the ultimate analysis be as follows:

	Per Cent
Carbon	74.79
Hydrogen	4.98
Oxygen	6.42
Nitrogen	1.20
Sulphur	3.24
Water	1.55
Ash	7.82
	100.00

When complete combustion takes place, as already pointed out, the carbon in the fuel unites with a definite amount of oxygen to form CO_2. The hydrogen, either in a free or combined state, will unite with oxygen to form water vapor, H_2O. Not all of the hydrogen shown in a fuel analysis, however, is available for the production of heat, as a portion of it is already united with the oxygen shown by the analysis in the form of water, H_2O. Since the atomic weights and H and O are respectively 1 and 16, the weight of the combined hydrogen will be $\frac{1}{8}$ of the weight of the oxygen, and the hydrogen available for combustion will be $H - \frac{1}{8}O$. In complete combustion of the sulphur, sulphur dioxide, SO_2, is formed.

Expressed numerically, the theoretical amount of air required for the above analysis is as follows: (See Column 6, Table 16.)

$$0.7479\ C \times 2\tfrac{2}{3} = 1.9944\ O$$

$$\left(0.0498 - \frac{0.0642}{8}\right) H \times 8 = 0.3262\ O$$

$$0.0324\ S \times 1 = 0.0324\ O$$

$$\text{Total required} \quad 2.3610\ O$$

One pound of oxygen is contained in 4.32 lb. of air.

The total air needed per pound of coal, therefore, will be $2.3610 \times 4.32 = 10.200$ lb.

The weight of combustible per pound of fuel is $0.7479 + 0.0418^* + 0.0324 + 0.012 = 0.83$ pounds, and the air theoretically required per pound of combustible is $10.200 / 0.83 = 12.3$ lb.

The above is equivalent to computing the theoretical amount of air required per pound of fuel by the formula: (See Column 8, Table 16.)

$$\text{Weight per pound} = 11.52\ C + 34.56\left(H - \frac{O}{8}\right) + 4.32\ S$$

where, C, H, O and S are proportional parts by weight of carbon, hydrogen, oxygen, and sulphur by ultimate analysis.

Theoretical and Actual Amount of Air Required. The calculations for air required presuppose that each and every particle of oxygen can be brought into intimate contact with the combustible. Practically this is impossible, due to the large amount of inert nitrogen present, variations in the fuel bed, and interference of clinker and ash, which cannot be removed as soon as formed. When burning oil and gas, however, some of these difficulties are eliminated, and the actual can more nearly approach the theoretical amount.

TABLE 17

THEORETICAL AMOUNT OF AIR REQUIRED

Fuel	Composition by Weight			Lbs. of Air per Lb. of Fuel
	% C	% H	% O	
Wood charcoal	93.0	....	....	11.16
Peat charcoal	80.0	....	....	9.6
Coke	94.0	....	....	10.8
Anthracite coal	91.5	3.5	2.6	11.7
Bituminous coal, dry	87.0	5.0	4.0	11.6
Lignite	70.0	5.0	20.0	8.9
Peat, dry	58.0	6.0	31.0	7.68
Wood, dry	50.0	6.0	43.5	6.00
Mineral oil	85.0	13.0	1.0	14.30

* Available hydrogen.

It is therefore necessary to provide for an *excess of air* when burning coal under either natural or forced draft, amounting to approximately 50 to 100 per cent of the net calculated amount, or about 18 to 24 lb. per pound of coal.

Less air results in *imperfect combustion* and smoke, while an excess cools the fire and setting and carries away large quantities of heat in the flue gases.

A table of the net calculated quantity of air per pound of fuel is given as a basis of comparison.

Rates of Combustion for Heating Boilers. Allowable combustion rates for varying sizes of grates are given in the following table:

TABLE 18

ALLOWABLE COMBUSTION RATES

Grate Areas	Coal per Sq. Ft. per Hour—Pounds	Remarks
6 sq. ft. or less, Small 6 – 10 sq. ft., Medium 10 sq. ft. or larger, Large	5 5.7 6.6	A variation of 10 per cent up or down from these rates is perfectly safe. The higher value for full sized chimneys with lined flues and the lower for unlined flues or long breeching connections.
4 to 8 sq. ft. 10 to 18 sq. ft. 20 to 30 sq. ft.	4 6 10	(*A. S. H. & V. E. Com. 1909.*) Rates of combustion reported for anthracite coal, as fired in internally fired heating boilers. See Transactions for further details.

The *fuel burning capacity of a square foot of grate,* where the chimney flue is of ample size, is controlled by the free area for air passage through the grate, by the area and length of the internal gas passages or flues in the boiler, and by the quantity and disposition of the heating or absorbing surface. The exact proportions can be arrived at only by an exhaustive series of experiments and tests.

A large grate area and an indifferent draft are a bad combination, because it is impossible to maintain good combustion over the entire area of the grate. If the combustion rate is much below six pounds, there may be a falling off of evaporative power due to an insufficient draft; and if the combustion rate much exceeds eight pounds, there may be a falling off in evaporative power due to the cooling influence of too much air.

Where the grate surface is too large, air is likely to mingle with the fuel in excess of requirements and cool the gases liberated by combustion. If there is an excess of flue surface (giving a long gas travel with the attendant frictional resistance) or too small a grate, insufficient air enters the fuel chamber, and the latent heat or stored energy of the fuel is not fully liberated.

A short boiler of a wide type is usually to be preferred, because it occupies less length in the pit, is more readily handled, its cost of erection is generally lower, and the fire being shorter is more readily controlled.

The *American Radiator Co.* state that the tests made by them for a period of eight years show a higher evaporative rate with a combustion rate of 8 lb. than with a rate of 4 lb. per sq. ft. of grate.

A series of six tests * on a round boiler gave the following results; half of the tests being run at the low rates and half at the higher rate:

Coal burned per sq. ft. of grate per hour	Water evaporated per lb. of coal burned
3.81 lb.	7.62 lb.
4.54	7.87
8.20	8.84

* NOTE.—These tests were run on a boiler having 65% direct and 35% indirect heating surface.

FLUE GAS ANALYSIS

Composition of Flue Gas. A flue gas analysis gives the proportion by volume of the principal constituent gases produced by the combustion of any fuel. The gases usually determined in such an analysis are carbon dioxide, CO_2, oxygen, O, and carbon monoxide, CO, while the residue or volume remaining after these gases are removed is taken as nitrogen, N.

By reference to Table 16 it will be seen that when oxygen and carbon combine the volume of the carbon dioxide gas formed is exactly equal to the volume of oxygen entering into the reaction, provided all volumes are measured at the same temperature and pressure. It, therefore, follows that if just sufficient air is provided to burn exactly one pound of pure carbon, the gas resulting will contain 20.91 per cent CO_2 and 79.09 per cent N, the oxygen having all entered into combination with the carbon, and the new gas resulting has simply taken the place of the original 20.91 per cent O. Now if 50 per cent excess air is supplied only ⅔ of the original oxygen volume will be replaced by CO_2 and the flue gas analysis will show 13.91 per cent CO_2, 7.0 per cent O and 79.09 per cent N. Finally, if 100 per cent excess air is supplied only ½ of the original oxygen volume will be replaced by CO_2 and the flue gas will contain 10.45 per cent CO_2, 10.45 per cent O, and 79.09 N. In each case the *oxygen* or *sum of the oxygen and carbon dioxide percentage is constant* or 20.91 per cent, while the *nitrogen percentage* is likewise *constant* at 79.09 per cent provided pure carbon only is burned completely.

If *carbon monoxide is produced* it will occupy twice the volume of the oxygen entering into its composition, hence the volume of the flue gas resulting will be greater (at the same temperature and pressure) than that of the air supplied by ½ of the per cent of CO present. One volume C + one volume O = two volumes CO.

If *hydrogen is present* in the fuel it will increase the apparent percentage of nitrogen in the flue gas, due to the fact that the water vapor formed by its combustion will condense at the temperature of the analysis, while the nitrogen brought in with the oxygen which combined with the hydrogen will remain as a gas and appear in the analysis.

Actual Air Supplied for Combustion. Likewise *the total* or *actual amount of air* supplied per pound of fuel burned can be expressed as follows, provided the flue gas analysis is known, and the relative densities of the gases are given.

These densities are in the same ratio as the molecular weights, which are as follows: CO_2 = 44, CO = 28, O_2 = 32, N_2 = 28; in which C = 12, O = 16 and N = 14.

In this connection it must be remembered that *equal volumes of all gases* at the same temperature and pressure *contain the same number of molecules*, hence the truth of the above statement.

It will therefore be apparent that if we let N_2, CO_2 and CO represent the percentages by volume from a flue gas analysis, and C_1 the percentage by weight of carbon in the fuel; then the pounds of air per pound of fuel will be expressed as follows:

$$A_a = \frac{28 \times N_2}{12(CO_2 + CO) \times 76.9} \times C_1,$$

where 76.9 = per cent of nitrogen in atmospheric air by weight and A_a = lb. of air supplied per pound of the fuel.

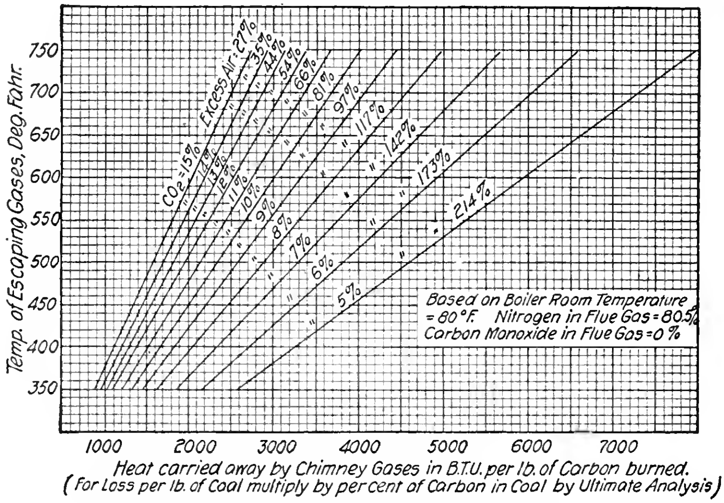
It should be noted that in the above expression all the carbon is supposed to burn and pass up the flue. Since this is never true in practice, it is necessary to correct C_1 by the amount of carbon in the ash. Thus, if the ash in a boiler test amounted to 16 per cent, and an analysis was found to contain 25 per cent of carbon, the percentage of unconsumed carbon would be $16 \times 0.25 =$ 4 per cent of the total coal burned. Now if the coal by ultimate analysis contained 80 per cent of carbon, only $80 - 4 = 76$ per cent of the fuel would actually be combustible carbon, hence use 76 per cent for C_1 in the above formula instead of 80 per cent, which is C_1, as reported in the analysis.

Then the *ratio of air actually supplied to that theoretically required* is A_a/A_t as determined above.

Weight of Flue Gas. The *weight of flue gases, W, per pound of carbon* is also easily computed from the flue gas analysis by the following formula,

$$W = \frac{44\ CO_2 + 32\ O_2 + 28(CO + N)}{12\ (CO_2 + CO)},$$

where the symbols CO_2, O_2, CO and N are the percentages by volume of these gases as determined from the flue gas analysis. Also the weight of flue gas per pound of dry coal may be de-



·Loss due to Heat carried away by Chimney Gases for Varying Percentages of Carbon Dioxide

FIG. 6.

termined from this formula by multiplying W by the percentage of carbon C_1 in the coal as found by an ultimate analysis.

Heat Lost in Flue Gas. The *heat lost in the flue gases due to the heat in the gases* is $L = 0.24\ W\ (t_2 - t_1)$ where L = B.t.u. lost per pound of dry coal, W = weight of flue gases per pound of dry coal, t_2 = temperature of flue gases, t_1 = temperature of air, and 0.24 = specific heat of the flue gases. The above loss is given graphically, as shown by Fig. 6, for varying percentages of CO_2 and different flue gas temperatures.

The *heat lost in the flue gases, due to the formation of carbon monoxide* when the carbon is incompletely burned is, in B.t.u. per pound of dry fuel,

$$L_1 = 10{,}150 \times \frac{12\ CO}{12(CO + CO_2)} \times C_1,$$

where 10,150 is the heat value per pound of carbon in the CO, and CO and CO_2 are percentages by volume from the flue gas analysis while C_1 is the proportion by weight of carbon which must be corrected to give the amount burned and passed up the stack as already explained.

Orsat Apparatus. The apparatus most commonly used for flue gas analysis is known as the *Orsat* (Fig. 7), and is described as follows:

"The burette A is graduated in cubic centimeters up to 100, and is surrounded by a water jacket to prevent any change in temperature from affecting the density of the gas being analyzed.

"For accurate work it is advisable to use four pipettes, B, C, D, E, the first containing a solu-

tion of *caustic potash for the absorption of carbon dioxide*, the second an alkaline solution of *pyrogallol for the absorption of oxygen*, and the remaining two an acid solution of *cuprous chloride for absorbing the carbon monoxide*. Each pipette contains a number of glass tubes, to which some of the solution clings, thus facilitating the absorption of the gas. In the pipettes *D* and *E*, copper wire is placed in these tubes to re-energize the solution as it becomes weakened. The rear half of each pipette is fitted with a rubber bag, one of which is shown at *K*, to protect the solution from the action of the air. The solution in each pipette should be drawn up to the mark on the capillary tube.

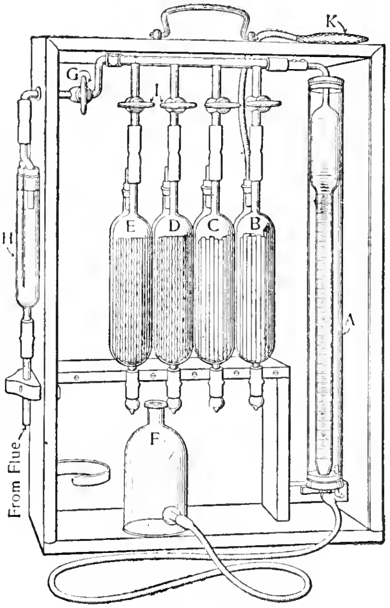


Fig. 7. Orsat Apparatus.

"The gas is drawn into the burette through the U-tube *H*, which is filled with spun glass, or similar material, to clean the gas. To discharge any air or gas in the apparatus, the cock *G* is opened to the air and the bottle *F* is raised until the water in the burette reaches the 100 cubic-centimeter mark. The cock *G* is then turned so as to close the air opening and allow gas to be drawn through *H*, the bottle *F* being lowered for this purpose. The gas is drawn into the burette to a point below the zero mark, the cock *G* then being opened to the air and the excess gas expelled until the level of the water in *F* and in *A* is at the zero mark. This operation is necessary in order to obtain the zero reading at atmospheric pressure.

"The apparatus should be carefully *tested for leakage* as well as all connections leading thereto. Simple tests can be made, as for example: If after the cock *G* is closed, the bottle *F* is placed on top of the frame for a short time and again brought to the zero mark, and the level of the water in *A* is above the zero mark, a leak is indicated.

"Before taking a final sample for analysis, the burette *A* should be filled with gas and emptied once or twice, to make sure that all the apparatus is filled with the new gas. The cock *G* is then closed and the cock *I* in the pipette *B* is opened and the gas driven over into *B* by raising the bottle *F*. The gas is drawn back into *A* by lowering *F* and when the solution in *B* has reached the mark in the capillary tube, the cock *I* is closed and a reading is taken on the burette, the level of the water in the bottle *F* being brought to the same level as the water in *A*. The operation is repeated until a constant reading is obtained, the number of cubic centimeters, absorbed as shown by the reading, being the percentage of CO_2 in the flue gases.

"The gas is then driven over into the pipette *C* and a similar operation is carried out. The difference between the resulting reading and the first reading gives the percentage of oxygen in the flue gases.

"The next operation is to drive the gas into the pipette *D*, the gas being given a final wash in *E*, and then passed into the pipette *C* to neutralize any hydrochloric acid fumes which may have been given off by the cuprous chloride solution, which, especially if it be old, may give off such fumes, thus increasing the volume of the gases and making the reading on the burette less than the true amount.

"The process must be carried out in the order named, as the pyrogallol solution will also absorb carbon dioxide, while the cuprous chloride solution will also absorb oxygen.

"As the pressure of the gases in the flue is less than the atmospheric pressure, they will not of themselves flow through the pipe connecting the flue to the apparatus. The gas may be drawn into the pipe in the way already described for filling the apparatus, but this is a tedious

method. For rapid work a rubber bulb aspirator connected to the air outlet of the cock G will enable a new supply of gas to be drawn into the pipe, the apparatus then being filled as already described. Another form of aspirator draws the gas from the flue in a constant stream, thus insuring a fresh supply for each sample.

"The analysis made by the *Orsat* apparatus is *volumetric*. If the *analysis by weight* is required, it can be found from the volumetric analysis as follows:

"Multiply the percentages by volume by either the densities or the molecular weight of each gas, and divide the products by the sum of all the products; the quotients will be the percentages by weight. For most work sufficient accuracy is secured by using the even values of the molecular weights."

Example. An application of the above data when an ultimate analysis of the fuel and a volumetric analysis of the flue gas is known can be made as follows:

Partial ultimate analysis, C = 82.1%, H = 4.25%, O = 2.6%, S = 1.6%, Ash = 6.0%, and B.t.u. per pound of dry Pocahontas coal = 14,500. The flue gas analysis is,

	Per Cent
CO_2	10.7
O	9.0
CO	0.0
N (by difference)	80.3

Determine: The flue gas analysis being given, (1) the amount of air required for perfect combustion, (2) the actual weight of air per pound of fuel, (3) the weight of flue gas per pound of coal, (4) the heat lost in the chimney gases if the temperature of these is 500° F., and (5) the ratio of the air supplied to that theoretically required.

Solution: The theoretical weight of air required for perfect combustion, per pound of fuel, from formula already given under "Air Required for Combustion," will be,

$$W_1 = 11.52 \times 0.821 + 34.56 \left(0.0425 - \frac{0.026}{8}\right) + 4.32 \times 0.016 = 10.88 \text{ lb.}$$

If the amount of carbon which is burned and passes away as flue gas is 80 per cent, which would allow for 2.1 per cent of unburned carbon in terms of the total weight of dry fuel burned, the weight of dry gas per pound of carbon burned will be from formula already given under "Weight of Flue Gas,"

$$W_2 = \frac{44 \times 10.7 + 32 \times 9.0 + 28\,(0 + 80.3)}{12\,(10.7 + 0)} = 23.42 \text{ lb.,}$$

and the weight of flue gas per pound of coal burned will be $0.80 \times 23.42 = 18.74$ lb.

The heat lost in the flue gases per pound of coal burned will be from formula and the value 18.74 just determined:

Loss $= 0.24 \times 18.74 \times (500-60) = 1{,}979$ B.t.u.

The percentage of heat lost in the flue gases will be $1{,}979 \times 100/14{,}500 = 13.6$ per cent.

The ratio of air supplied per pound of coal to that theoretically required will be $(18.74 - 1)/10.88 = 1.63$.

The ratio of air supplied per pound of combustible to that required will be,

$$\frac{0.803}{0.803 - 3.782\,(.09 + \tfrac{1}{2} \times 0)} = 1.73$$

$$\text{since, } \frac{N}{N - 3.782\,(O + \tfrac{1}{2}\,CO)} = \frac{\%\text{ nitrogen in whole amount of air.}}{\%\text{ nitrogen in air actually required.}}$$

Note. The value 3.782 is the volumetric ratio of nitrogen to oxygen in the air (Table 16, Column 17). All the uncombined oxygen and ½ of the carbon monoxide represents the oxygen equivalent of unnecessary or excess nitrogen, which in turn represents air.

The ratio based on combustible will be greater than the ratio based on fuel if there is unconsumed carbon in the ash.

Unreliability of CO_2 Readings Taken Alone. It is generally assumed that high CO_2 readings are indicative of good combustion and hence of high efficiency. This is true only in the sense that such high readings do indicate the small amount of excess air that usually accompanies good combustion, and for this reason high CO_2 readings alone are not considered entirely reliable. Wherever an automatic CO_2 recorder is used, it should be checked from time to time and the analysis carried further with a view to ascertaining whether there is CO present. As the percentage of CO_2 in these gases increases, there is a tendency toward the presence of CO, which, of course, cannot be shown by a CO_2 recorder, and which is often difficult to detect with an Orsat apparatus. The greatest care should be taken in preparing the cuprous chloride solution in making analyses and it must be known to be fresh and capable of absorbing CO.

Smokeless Combustion. Smokeless combustion can only be attained with special equipment and most careful firing, and the following methods for its accomplishment are recommended by the *Babcock & Wilcox Co.*, who have had a wide experience in this field:

"The question of smoke and smokelessness in burning fuels has recently become a very important factor in the problem of combustion. Cities and communities throughout the country have passed ordinances relative to the quantities of smoke that may be emitted from a stack, and the failure of operators to live up to the requirements of such ordinances, resulting as it does in fines and annoyance, has brought their attention forcibly to the matter.

"The whole question of smoke and smokelessness is to a large extent a comparative one. There are any number of plants burning a wide variety of fuels in ordinary hand-fired furnaces, in extension furnaces and on automatic stokers that are operating under service conditions, practically without smoke. It is safe to say, however, that *no plant will operate smokelessly under all conditions of service*, nor is there a plant in which the degree of smokelessness does not depend largely upon the intelligence of the operating force.

"When a condition arises in a boiler room requiring the fires to be brought up quickly, the operatives in handling certain types of stokers will use their slice bars freely to break up the green portion of the fire over the bed of partially burned coal. In fact, when a load is suddenly thrown on a station the steam pressure can often be maintained only in this way, and such use of the slice bar will cause smoke with the very best type of stoker. In a certain plant using a highly volatile coal and operating boilers equipped with ordinary hand-fired furnaces, extension hand-fired furnaces and stokers, in which the boilers with the different types of furnaces were on separate stacks, a difference in smoke from the different types of furnaces was apparent at light loads, but when a heavy load was thrown on the plant, all three stacks would smoke to the same extent, and it was impossible to judge which type of furnace was on one or the other of the stacks.

"In *hand-fired furnaces* much can be accomplished by proper firing. A combination of the alternate and spreading methods should be used, the coal being fired evenly, quickly, lightly, and often, and the fires worked as little as possible. Smoke can be diminished by giving the gases a long travel under the action of heated brickwork before they strike the boiler heating surfaces. Air introduced over the fires and the use of heated arches, for mingling the air with the gases distilled from the coal will also diminish smoke. Extension furnaces will undoubtedly lessen smoke where hand-firing is used, due to the increase in length of gas travel, and the fact that this travel is partially under heated brickwork. Where hand-fired grates are immediately under the boiler tubes, and a highly volatile coal is used, if sufficient combustion space is not provided, the volatile gases, which are distilled as soon as the coal is thrown on the fire, strike the tube surfaces and are cooled below the burning point before they are wholly consumed and therefore pass through as smoke. With an extension furnace, these volatile gases are acted upon by the radiant heat from the extension furnace arch, and this heat, together with the added length of travel, causes their more complete combustion before striking the heating surfaces than in the former case.

"Smoke may be diminished by employing a baffle arrangement which gives the gases a fairly long travel under heated brickwork and by introducing air above the fire. In many cases, how-

ever, special furnaces for smoke reduction are installed at the expense of capacity and economy.

"From the standpoint of smokelessness, undoubtedly the best results are obtained with a good stoker, properly operated. As stated above, *the best stoker will cause smoke under certain conditions.* Intelligently handled, however, under ordinary operating conditions, stoker-fired furnaces are much more nearly smokeless than those which are hand-fired, and are, to all intents and purposes, smokeless. In practically all stoker installations there enters the element of time for combustion, the volatile gases as they are distilled being acted upon by ignition or other arches before they strike the heating surfaces. In many instances, too, stokers are installed with an extension beyond the boiler front, which gives an added length of travel, during which, the gases are acted upon by the radiant heat from the ignition or supplementary arches, and here again we see the long travel giving time for the volatile gases to be properly consumed.

"Finally, it must be emphatically borne in mind that the question of smokelessness is largely one of degree, and dependent to an extent much greater than is ordinarily appreciated upon the handling of the fuel and the furnaces by the operators, be these furnaces hand-fired or automatically fired."

CHAPTER VI

STEAM HEATING BOILERS AND HOT-WATER HEATERS

BOILER PERFORMANCE

Pressures, Attention and Materials. Heating boilers usually operate under much *lower pressures* than do power boilers, and in most cases receive far less attention. The steam boilers are usually designed to operate on from *2 to 5 lb.* steam pressure, and the water " boilers " or hot-water heaters are seldom subjected to a hydrostatic head in excess of *100 ft.* when in operation.

The *attention* given these boilers is of such an *intermittent character* that they must carry the heating load for comparatively long periods without firing. These periods may range from 6 to 10 hours and in consequence the combustion rate is low, and relatively large grates and fire pots are necessary.

The *materials* employed for constructing heating boilers are *cast iron*, especially for the smaller sizes, although boilers of nearly 100 equivalent steam boiler horsepower (see Ratings) are made of this same material; and *steel* or *wrought iron* which are more generally used in the larger sizes. The government departments usually specify steel heating boilers, and they are used extensively in office and loft buildings as well.

Boiler Heating Surface. The *capacity* of any boiler or water heater depends on the amount of, and the temperatures on the opposite sides of, the heat transmitting surfaces in contact with the water in the boiler on one side, and the fire or hot gases on the other. It is most important that a rapid circulation of water and the hot gases shall take place over these surfaces, and preferably in opposite directions.

Two kinds of surface are distinguished in boiler practice, and known as direct and indirect surface. *Direct surface* is that on which the fire shines, and *indirect* that in contact with the flue gases only. All such surface must have water on the opposite side.

In some boilers the hot gases are allowed to come in contact with the boiler surface above the water line so that there is only steam in contact with this surface on the inner side. Such surface is known as *superheating surface* in order to distinguish it from ordinary heating surface.

Direct surface is the more valuable of the two, per sq. ft., as it is usually subjected to a higher temperature, and furthermore because the intensity of radiation from an incandescent surface appears to vary as some power of the temperature of that surface, either the third or fourth. See *Stefan Boltzman* Law in chapter on " Measurement of Heat."

The *amount of heat absorbed*, as given by *R. C. Carpenter*, by the boiler heating surface depends upon the circulation of the heated gases, the circulation of the water, and the difference in temperature between the gases and the water. The average absorption in power boilers varies between 2000 and 3000 B.t.u. per sq. ft. per hour, and the ratio of grate to heating surface varies between 1 to 40 and 1 to 60. With house heating boilers, either for water or steam it is probably not desirable in the interests of economy to require a heat absorption exceeding 1800 to 2400 B.t.u. per sq. ft. per hour. As each sq. ft. of radiating surface requires about 250 B.t.u. per hour for steam, and 150 B.t.u. for hot water, one square foot of heating surface would, under these conditions, supply from 7.2 to 9.6 sq. ft. of radiating surface for steam, and 12 to 16 sq. ft. for hot water.

Equivalent Evaporation. The equivalent evaporation of a boiler is the pounds of water the boiler would evaporate per pound of coal burned if it received the feed water at 212°, and evaporated it into steam at this same temperature and pressure, so that the evaporation would take place *from and at 212° F.* In practice the feed water is usually below this temperature and

evaporation actually takes place at some higher temperature than 212°. Hence, to find the *equivalent evaporation* it is always necessary to make use of the following relation:

$E = \dfrac{(x_2 r_2 + q_2 - q_1)}{971.7} \times P$, where the fractional part of the expression is known as the *factor of evaporation*, so that $E =$ Factor of Evaporation $\times P$.

E = Equivalent evaporation *from and at 212° F.* in pounds.
x_2 = Quality of steam as actually evaporated.
r_2 = Latent heat of steam as actually evaporated.
q_2 = Heat of the liquid as actually evaporated.
q_1 = Heat of the liquid as actually fed to boiler.
P = Actual evaporation in pounds per pound of fuel burned.
971.7 = Latent heat of steam at 212° F.

Boiler Horsepower. A boiler horsepower is the energy required to evaporate 34.5 lb. of water at 212°F. into *dry* steam at 212°F. The horsepower rating of a boiler is always measured in terms of the equivalent evaporation. Thus, if we divide the equivalent evaporation of a boiler by 34.5 we get the boiler horsepower developed.

Boiler Efficiences. Heating boilers, operated at their rated capacity, will show an efficiency of from 55 to 65 per cent. This efficiency is the ratio of heat absorbed per pound of dry coal by the water and steam in the boiler to the actual heat value of one pound of the coal, and is *the combined efficiency* of the boiler and furnace.

The following table shows at a glance the pounds of water a boiler must evaporate *from and at 212° F.* per pound of dry coal fired, for coals of varying calorific value, in order to give various combined efficiencies.

TABLE 1

EQUIVALENT EVAPORATION PER POUND OF COAL*

	Combined Boiler and Furnace Efficiencies													
	50%		55%		60%		65%		70%		75%		80%	
B.t.u. per Lb. Dry Coal	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.	Evaporation per Lb. Coal	Lb. Coal per Boiler H.P. Hr.
7500	3.883	8.88	4.272	8.08	4.660	7.40	5.048	6.84	5.436	6.35	5.825	5.92	6.213	5.56
8000	4.142	8.33	4.556	7.58	4.971	6.94	5.385	6.40	5.799	5.96	6.213	5.56	6.627	5.20
8500	4.453	7.75	4.898	7.05	5.343	6.46	5.789	5.96	6.234	5.54	6.679	5.16	7.124	4.84
9000	4.660	7.41	5.126	6.73	5.592	6.18	6.058	5.70	6.524	5.28	6.990	4.94	7.456	4.63
9500	4.919	7.02	5.411	6.38	5.902	5.84	6.394	5.40	6.886	5.01	7.378	4.68	7.870	4.38
10000	5.224	6.60	5.747	6.00	6.269	5.50	6.791	5.08	7.314	4.72	7.836	4.41	8.359	4.13
10500	5.436	6.35	5.980	5.78	6.524	5.28	7.067	4.88	7.611	4.54	8.155	4.23	8.698	3.97
11000	5.695	6.06	6.245	5.52	6.834	5.05	7.404	4.66	7.973	4.33	8.543	4.04	9.113	3.79
11500	5.954	5.80	6.550	5.26	7.145	4.83	7.740	4.46	8.336	4.14	8.931	3.86	9.527	3.62
12000	6.213	5.56	6.834	5.05	7.456	4.63	8.077	4.27	8.698	3.97	9.320	3.70	9.941	3.47
12500	6.472	5.34	7.119	4.85	7.767	4.44	8.415	4.10	9.062	3.82	9.709	3.55	10.356	3.32
13000	6.731	5.12	7.404	4.66	8.077	4.27	8.750	3.94	9.423	3.66	10.096	3.42	10.769	3.20
13500	6.990	4.94	7.689	4.49	8.388	4.11	9.087	3.80	9.786	3.55	10.485	3.29	11.184	3.09
14000	7.249	4.76	7.973	4.33	8.698	3.97	9.423	3.66	10.148	3.40	10.873	3.17	11.598	2.98
14500	7.508	4.60	8.258	4.18	9.009	3.83	9.760	3.53	10.511	3.28	11.261	3.06	12.012	2.87

* Note.—The values in this table were calculated using the value 965.7 for the latent heat of steam at 212° F.

Efficiencies at head of columns are combined boiler and furnace efficiencies. Evaporation is given in pounds of water *from and at 212° F.* per pound of dry coal. The heating value

can also be taken for coal *as fired* whence the *evaporation per pound of coal* and the "pounds coal per boiler h.p. hr." will also be referred to coal "as fired." This latter use of the table will contain a slight error due to the absorption of a portion of the heat by the moisture in the coal.

$$\text{b.h.p.} = 34.5 \times 971.7 = 33{,}524 \text{ B.t.u.}$$

Example. Since the steam from 1 lb. of water will supply 4 sq. ft. of standard direct cast-iron radiation it follows that if the rate of combustion, calorific value of the coal, and the efficiency of a heating boiler are known, its rating can be found from this table, as follows: Given a boiler having a grate area of 13.68 sq. ft. and allowing a combustion rate of 7 lb. of coal of 12,000 B.t.u. heat value at 60 per cent efficiency we have from table an evaporation = 7.456 lb. Hence the rating = 13.68 × 7 × 7.456 × 4 = 2,860 sq. ft. of standard steam radiation.

RATING OF HEATING BOILERS

Standard Conditions. It is the general custom of American manufacturers of heating boilers to rate their boilers in terms of the number of square feet of standard direct cast-iron radiating surface which the boiler is capable of supplying under the following conditions:

Steam Boilers—steam pressure 2 lb. gage at boiler.

Hot Water Boilers—water temperatures: 180° F. leaving, and 160° F. entering boiler.

Fuel—anthracite coal of stove size.

The rate of combustion, see chapter on "Fuels and Combustion," or amount of coal necessary per hour for the boiler to develop its rating has, until recently, seldom been given, and the method of determining the rating has varied with different makers, and is seldom stated. Moreover, it is possible for a boiler to be placed on the market and assigned a certain rating although such rating has never been actually checked by test.

It therefore becomes most important to not only establish standard conditions for rating tests, but to require the manufacturer to be in a position to produce certified test sheets of such tests for his complete line of boilers.

The *standard conditions* under which a boiler should be tested to develop its rating are generally understood by the manufacturers at the present time to be as follows:

(*a*) Pressure, temperature and fuel as stated above.

(*b*) Fuel capacity to be sufficient to carry the boiler from 6 to 8 hrs. on one charge and leave 20 per cent reserve for igniting fresh charge.

(*c*) Draft of sufficient intensity to burn the fuel at the required rate.

(*d*) Each sq. ft. of direct cast-iron radiation has a transmission value of 250 B.t.u. and 150 B.t.u. per hour for steam and water radiators respectively.

(*e*) The condensation from steam radiators returns to the boiler at the same temperature as the steam or without loss of heat, so that the boiler simply supplies the latent heat of evaporation at 2 lb. pressure.

(*f*) The water from hot water radiators returns to the boiler at 160° allowing a 20° drop in the radiator, so that there is no loss in temperature allowed in the return main.

(*g*) Suitable heat allowance must be made for all connecting piping and boiler surface, and such surface must be figured as radiating surface or its equivalent. A general rule is to add, for an ordinary installation, about 25 per cent of the sq. ft. of radiation installed, in calculating the total load on the boiler.

Efficiency and Rate of Combustion as a Basis for Rating Heating Boilers. A simple method of *rating heating boilers* can be readily devised if the *efficiency* of the boiler and *rate of combustion* are known. The following table forms the basis for this method, and shows the range over which ratings may vary, for varying efficiencies and rates of combustion. The table can also be used to find size of grate if total amount of radiation or its equivalent is known.

Example. (See Table 2.) If 4 lb. coal per sq. ft. of grate per hour are burned under a boiler, the efficiency of which is 60 per cent, with 6 sq. ft. of grate, what will be the capacity of the boiler? From

the table, the radiation surface per sq. ft. of grate = 115.2 sq. ft., so that the total for 6 sq. ft. of grate = 691 sq. ft. of direct steam radiation. Further use of Table 2 to check manufacturer's ratings is shown below.

TABLE 2

RATINGS OF CAST IRON BOILERS IN TERMS OF SQUARE FEET OF DIRECT STEAM RADIATION PER SQUARE FOOT OF GRATE AREA, WITH DIFFERENT RATES OF COMBUSTION AND DIFFERENT BOILER EFFICIENCIES

(*Report of Committee on Rating Heating Boilers, Annual Meeting* 1911—*A. S. H. & V. Engrs.*)

Assumptions.—(a) Coal heat value = 12,000 B.t.u. per pound; (b) boiler efficiency = ratio of heat given off beyond nozzle to heat-value of coal burned; (c) one square foot of direct steam radiating surface or its equivalent gives off 250 B.t.u. per hour.

Lb. of Coal per Sq. Ft. of Grate per Hr.	Boiler Efficiencies										
	% 50.0	% 52.5	% 55.0	% 57.5	% 60.0	% 62.5	% 65.0	% 67.5	% 70.0	% 72.5	% 75.0
	Square Feet of Direct Radiation per Square Foot of Grate Area										
1	24.0	25.2	26.4	27.6	28.8	30.0	31.2	32.4	33.6	34.8	36.0
2	48.0	50.4	52.8	55.2	57.6	60.0	62.4	64.8	67.2	69.6	72.0
3	72.0	75.6	79.2	82.8	86.4	90.0	93.6	97.2	100.8	104.4	108.0
4	96.0	100.8	105.6	110.4	115.2	120.0	124.8	129.6	134.4	139.2	144.0
5	120.0	126.0	132.0	138.0	144.0	150.0	156.0	162.0	168.0	174.0	180.0
6	144.0	151.2	158.4	165.6	172.8	180.0	187.2	194.4	201.6	208.8	216.0
7	168.0	176.4	184.8	193.2	201.6	210.0	218.4	226.8	235.2	243.6	252.0
8	192.0	201.6	211.2	220.8	230.4	240.0	249.6	259.2	268.8	278.4	288.0
9	216.0	226.8	237.6	248.4	259.2	270.0	280.8	291.6	302.4	313.2	324.0
10	240.0	252.0	264.0	276.0	288.0	300.0	312.0	324.0	336.0	348.0	360.0

NOTE.—All radiating surfaces giving off different amounts of heat than 250 B.t.u. per hour per square foot may be reduced to "equivalent direct surface" at 250 B.t.u. per hour per square foot for use in connection with this table.

Example. A certain boiler has 18 sq. ft. grate area and is rated at 5,275 sq. ft. of radiation. Now 5,275/18 = 293 sq. ft. per 1 sq. ft. grate. Refer to above table and find a combustion rate of 9 lb. at 67.5 per cent efficiency. Another boiler has 6.12 sq. ft. grate area and is rated at 1,200 sq. ft. radiation. Again, 1,200/6.12 = 196 sq. ft. per 1 sq. ft. grate. Refer to above table and find a combustion rate of 6 lb. at 67.5 per cent efficiency. It is desired to know the grate area of first boiler with an 8 lb. combustion rate and 62.5 per cent efficiency. Refer to table, and find 240 sq. ft. of radiation per 1 sq. ft. grate. Then 5,275/240 = 22 sq. ft. of grate.

All of the above examples are based on anthracite coal of 12,000 B.t.u. per 1 lb. For coals of higher or lower heat values modify the table values by the ratio of the calorific values, at the proper efficiency for the other coal.

TABLE 3

AVERAGE RATES OF COMBUSTION FOR HEATING BOILERS (SAFE INCREASE 10 PER CENT)

Boiler Size	Small	Medium	Large
Grate, square feet	6 or less	6 – 10	10 or over
Coal per hour per square foot	5 lb.	5.7 lb.	6.6 lb.
Run on one firing with 20 per cent reserve	8 hr.	7 hr.	6 hr.

NOTE.—Boilers with grates above 20 sq. ft. can maintain a combustion rate of from 10–15 lb. per sq. ft. and are usually operated under power boiler conditions—constant attendance. See also chapter on "Fuels and Combustion."

Equivalent Boiler Horsepower as a Basis for Rating Heating Boilers. The capacities of heating boilers may be stated in boiler horsepower, and the equivalent of same in square feet of standard radiation may be easily determined as follows:

Since 1 boiler horsepower is equal to 34.5 lb. of water evaporated per hour, from and at 212° F., the boiler must deliver 34.5 × 971.7 (latent heat at 212° F.) = 33,524 B.t.u. per hour. Now since 1 sq. ft. of standard cast iron steam radiation transmits 250 B.t.u. per hour, one

boiler horsepower equals 33,524 / 250 = 134.1 sq. ft. of this radiation, or 1 sq. ft. of direct cast iron steam radiation = 0.00756 boiler horsepower.

It also follows that the equivalent boiler horsepower rating of a hot water heater is 33,524/ 150 = 223.5 sq. ft. of direct cast-iron hot water radiation, or 1 sq. ft. of direct cast-iron hot water radiation = 0.00447 boiler horsepower.

From the above relation it will be apparent that *one boiler horsepower will supply steam for 100 sq. ft. of direct cast-iron steam radiation* and allow, for secondary losses from mains, risers, and returns, a margin of 34.1 per cent, which justifies the common practice of allowing one boiler horsepower for each 100 sq. ft. of low pressure steam radiation installed.

Grate Surface. It is always advisable to check *the grate area required* for heating boilers, especially if the total heat loss to be supplied by the boiler is known. This total heat loss must include not only the calculated loss, due to transmission through walls and glass, for which the radiation is proportioned, but also about 20 to 30 per cent additional for heat losses from the piping system and boiler. So that, if H is the building loss in B.t.u., we have 1.25 H = total B.t.u. loss.

Then
$$G = \frac{1.25\ H}{C \times F \times E}$$

where C = rate of combustion in pounds of dry coal per sq. ft. of grate area per hour, F = calorific value of fuel in B.t.u. per pound of dry coal, and E = the combined efficiency of boiler and grate. G will be in sq. ft. and the boiler selected should have not less than this grate area. Special attention is called to the distinction between grate area and firebox or fuel-pot area as explained below under "Depth of Fuel Pot."

Depth of Fuel Pot. The average of the firebox is usually somewhat larger than the grate area in sectional boilers, while it may be less than the grate area in certain types of round boilers. In any event the capacity of the firebox or fuel pot from grate to middle of fire door should always be sufficient to hold all the coal required for an 8-hour firing period, plus at least 20 per cent reserve to be used for igniting a fresh charge.

The following method is used to determine the depth of pot or the firing period as the case may be. Let G = grate area in sq. ft., C = rate of combustion, A = average area of firepot, h = firing period in hours, W = weight of fuel per cu. ft. (50 lb. for anthracite and 40 lb. for bituminous), D = depth of fuel bed in ft. Then ($G\ C\ h$) + 20 per cent (allowance to ignite fresh charge) = total weight of one charge, also $A\ W\ D$ = total weight of one charge, hence

$$D = \frac{1.2\ G\ C\ h}{A\ W}, \text{ or } h = \frac{A\ W\ D}{1.2\ G\ C}$$

As noted above D is measured from grate to center of fire door, which varies from 8″ x 14″ in small, to 11″ x 19″ in large boilers. This formula allows for the greater bulk of soft coal.

Example. Given a boiler with grate area of 8 sq. ft., average area firepot 9 sq. ft., height to center of fire door = 18 in., rate of combustion = 6 lb. per sq. ft. of grate for anthracite coal. Required the number of hours this boiler will carry its load on one charging. $h = \frac{9 \times 50 \times 1.5}{1.2 \times 8 \times 6} = 11.7$ hours.

Effect of Fuels on Ratings. All *ratings are based on anthracite coal of stove size* unless otherwise stated. In case bituminous coal is used and boiler is selected by catalog rating, a boiler with firepot having at least 25 per cent greater capacity should be selected for the same weight of coal occupies 25 per cent space more. With *soft coal* additional heating surface is also required as the accumulation of soot from such coal renders the heating surfaces less effective than when hard coal is used.

Boilers for *pea coal* should also have a larger firepot than those for stove or furnace coal. The *small sizes of anthracite* contain far more ash than the larger sizes, and hence have a greater bulk for the same heating effect, so that larger fuel-pots for the same capacity will be required.

Firing periods differing from the one on which the boiler is rated will also affect the fuel-holding capacity. For instance, should it be desired to operate a certain line of boilers designed for an 8-hour period on a 12-hour basis at least 50 per cent greater fuel holding capacity will be necessary and a larger boiler must be selected, as shown by formula already given for depth of fuel pot.

Fuel Consumption. The fuel consumption of heating boilers per season will be found under the chapter on "Fuels and Combustion."

Equivalent Rating for Conditions other than Standard. It often happens that the load connected to a steam or hot water boiler may not be operated under the standard conditions previously assumed as a basis of rating. In this case tables of ratings cannot be used until *the equivalent value of this load* in terms of sq. ft. of standard cast iron radiation has been determined.

The following relations show a method for finding such equivalent values.

Let R = sq. ft. standard cast-iron radiation = 250 B.t.u. per sq. ft. for steam, and 150 B.t.u. per sq. ft. for water. Also let:

r = actual sq. ft. of radiation to be supplied.
K = coefficient of transmission for this radiation.
t_s or t_w = temperature of steam or average temperature of hot water in the radiator.
t_a = temperature of air surrounding radiator.
$K(t_s - t_a)$ = radiation factor or B.t.u. given off per sq. ft. per hr.

$$R_s = \frac{r_1 \times K_1 (t_s - t_a)}{250} \text{ and } R_w = \frac{r_2 \times K_2 (t_w - t_a)}{150.}$$

Example. (Steam heating.) Required the size of boiler (rating in sq. ft. of standard cast-iron radiation) to supply 1,000 sq. ft. of direct pipe coil radiation. Steam pressure = 5-lb. gage. Air = 65° F. K by test = 2.42 B.t.u. From steam tables t_s = 227.14, $R = 1{,}000 \times \frac{2.42\ (227.14 - 65)}{250} = 1{,}000 \times \frac{2.42 \times 162.14}{250} = 1{,}570$ sq. ft. To this add 25 per cent for mains and risers, or, 1.25 × 1,570 = 1,962 sq. ft., or practically a 2,000 sq. ft. capacity boiler, will be required.

Grate should be checked by calculation previously given to ascertain minimum size. $G = \frac{1.25\,H}{C \times F \times E}$

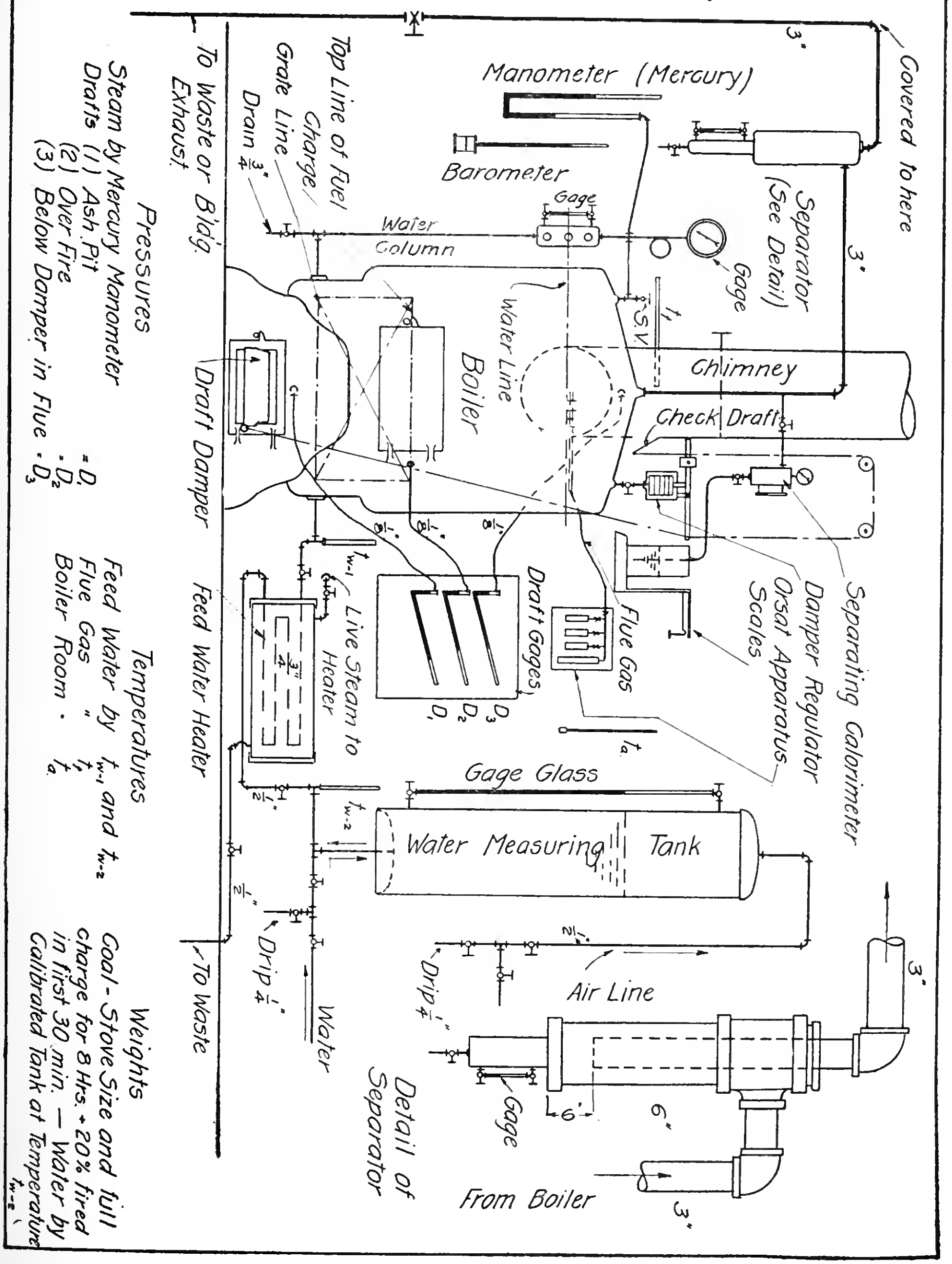
Example. (Water heating.) Let Q = Total number of gals. of water to be heated in h hours. $W = \frac{8\frac{1}{3} \times Q}{h}$ = weight of water to be heated per hour, t_1 = initial temperature of water, t_2 = final temperature of water. Then $W(t_2 - t_1)$ = B.t.u. to be supplied per hour. Hence $W(t_2 - t_1)/150$ = hot water heater rating required. $W(t_2 - t_1)/250$ = steam boiler rating required.

Example. A swimming pool contains 50,000 gals. of water, and this water is heated by being passed through a hot water heater in four hours. Entering temperature = 50° F. and final temperature = 75. F°

Hot water radiation reduced to equivalent standard value $= \frac{50{,}000 \times 8\frac{1}{3}}{(4 \times 150)} \times (75\text{–}50) = 17{,}350$ sq. ft. = rating of hot water heater to which must be added any losses from piping, etc.

Tests of Heating Boilers. It is absolutely necessary, as already stated, that complete and accurate evaporative and water-heating tests be run on heating boilers before any attempt is made to give them a rating. The larger manufacturers have established testing plants in which each line of boilers is tested for performance under working conditions before any ratings are determined upon or published. These ratings should always be less than the results obtained in the plant tests to allow for unfavorable conditions existing in the actual installation and operation of the boiler.

Fig. 1 Heating Boiler Test - Manufacturer's Rating



Manufacturer's Rating Test. The apparatus for conducting a heating boiler test for the determination of its proper rating as carried out at the *University of Illinois* is shown in Fig. 1. The method employed consists in bringing the boiler to steam pressure (2 lb. gage) and temperature with a preliminary fire which is then drawn, and a known weight of wood fired at the instant of starting the actual test. This is then followed by the entire charge of stove size anthracite coal at two fifteen minute intervals, so that all fuel plus 20 per cent rekindling reserve is fired within 30 minutes.

Water is fed, from calibrated measuring tanks at a temperature of 180° F. Compressed air is used to force this water through the feed water heater at a uniform rate of supply, and maintain a constant water line.

Pressures are maintained constant by the use of a damper regulator which can be made to operate the drafts in such a way as to keep the water level constant by simply varying the opening of the gate valve on the discharge line.

Temperatures, drafts, and quality of steam are read as indicated in the figure, and a careful record of all water entrained or otherwise thrown over into the separator is made.

The apparatus is found to be entirely reliable in duplicating results on consecutive runs or tests. The flue gas temperatures must be closely watched at the start until conditions become constant, and the rate of feed retarded somewhat until all the coal is ignited.

The test is continued for 6, 8, or 10 hours, during which time *the fire is not touched or disturbed in any way,* depending on the capacity to be developed and the firing period. The fire is then drawn and killed by the use of a small amount of water, and the residual coal and ash separated and weighed. This coal is analyzed or estimated in terms of equivalent good coal, and must be within 15 to 25 per cent of the coal actually burned. The evaporation is then corrected for a residual coal balance of exactly 20 per cent on which the boiler rating is based, using 0.25 lb. of steam *from and at 212°* per sq. ft. of standard radiation allowed for the rating.

It is of course necessary to know or ascertain the approximate capacity of the boiler in advance.

This plant is also adapted to *tests with soft coal,* but in that case periodic firing is required, depending on the fuel used, and the *test is not a rating test.*

Data from Rating Tests with Anthracite Coal. A sample manufacturer's test log sheet follows and indicates the amount and character of the data this manufacturer ascertains as a *basis for rating heating boilers.* This sheet has been somewhat abridged, but all essential items are included, and the notes appended marked *A*, *B*, *C*, and *D* will explain the significance of the various items.

Results of tests made by *The American Radiator Company* at their Institute of Thermal Research on a No. S–48–7 Ideal Sectional Steam Boiler, the catalog rating of which is placed at 5,700 square feet of direct radiation (including radiators, mains, and risers):

Size of grate, 48 × 64½ in. grate area			21.6 sq. ft.
Heating surface—total			300 " "
1 Fuel used in tests	Hard coal	Hard coal	Hard coal
2 No. of boiler	S–48–7	S–48–7	S–48–7
3 Duration of test—hours	8	7	8
4 Fuel burned during tests—lb	1,360.00	1,344.00	1,434.00
5 Fuel per hour—lb	170.00	192.00	178.20
6 Fuel per sq. ft. grate per hour—lb	7.90	8.95	8.35
7 Stack temperature—degrees Fahr	750.00	725.00	600.00
8 Evaporation per sq. ft. of heating surface per hour—lb	4.97	5.60	5.24
9 Evaporative power available—lb. water per lb. of coal	8.80	8.75	8.77
10 Boiler-power (evaporation per hour)—lb. (item 5 × item 9)	1,495.00	1,680.00	1,562.00
11 Capacity—sq. ft. (item 10/0.22)	6,800.00	7.640.00	7.100.00

12 Capacity—sq. ft. (item 10/0.25)	5,980.00	6,720.00	6,250.00
Catalog rating			5,700 sq. ft.

Note A. The three tests in this summary show the effect of different rates of combustion, as noted in items 5, 6, 7, 8, and 9.

Note B. Item 10 is called the boiler-power because it is the quantity of heat available per hour expressed in pounds of water evaporated by the hourly consumption of fuel. The condensation from a large number of regular working installations has been measured, and it is found that the average condensation of the radiation and piping (when the latter is covered) is 0.22 pound of steam per square foot per hour. Dividing items on line 10 by 0.22 gives the capacities shown in item 11. But a general average of experiences has justified a more liberal condensation rate of 0.25 lb. of steam per hr. which has been established as the basis for catalog ratings of 48-inch boilers.

Note C. The capacity of boilers of this type, which have a regular fireman's care, is not figured on the eight-hour, coal-available basis. The hourly boiler-power, item 10, may be maintained indefinitely, or as long as the rate of combustion is continued.

In most of the above tests the boiler was coaled at intervals of three to five hours, thus conforming as nearly as possible to current practice in the operation of boilers of this large size.

These tests show the fuel which must be burned per hour to maintain a steam pressure of 2 pounds on every square inch of radiation. In actual operation just about one-third of these quantities is burned, for the reason that the maximum demand is for short periods only.

Note D. *Method of Calculating Radiation Capacity.* The catalog rating is based on burning 167.5 pounds of coal per hour (a rate of combustion needed only in extremely cold weather). The evaporative rate is 8.5 pounds of water per pound of coal, which all tests amply justify (item 9) and gives the rated capacity:

$$\frac{167.5 \times 8.5}{0.25} = 5{,}700 \text{ sq. ft.}$$

The test in Column 2, based on the burning of 192 pounds of coal per hour, estimated on a condensation rate of 0.25 pounds, developed a capacity of 6,720 square feet.

$$\frac{192 \times 8.75}{0.25} = 6{,}720. \text{ sq. ft.}$$

This test shows that a small increase in coal burned per hour increases the boiler's capacity over 740 square feet more than was developed in test in Column 1.

Tests with Bituminous Coal. Rating tests are not made with bituminous coal, as it is too variable in its composition and heat value, and requires frequent firing and attention.

The tests made at the *University of Illinois* on small cast-iron house heating boilers of 1,200 to 1,400 sq. ft. capacity show efficiencies of 51.70 per cent for Illinois lump coal, and 57.42 per cent for Pocahontas coal.

The use of this coal results in the production of a large amount of smoke and soot, so that the efficiency of the boiler heating surfaces is seriously affected.

In order to overcome the smoke nuisance and loss of efficiency in burning soft coal directly, boilers with down-draft grates are coming into more general use, and the following comparative tests on brick set steel firebox boilers have been compiled from reports by *Robert W. Hunt & Co.* for the *Kewanee Boiler Co.*, using both plain and down-draft grates.

Tests of Firebox Heating Boilers. *Object of Tests.* To obtain the relative efficiencies of the boiler as to fuel economy and smoke prevention, and also to observe the relative operation of the same boiler using five different kinds and grades of bituminous coal.

Boilers. (See Figs. 18 and 19.) Two Kewanee Firebox Steam Boilers of the same general design, set in brickwork, were used. The products of combustion after passing through the

fire-tubes were returned around the lower part of the shell and around the firebox to a stack set over the front end of the installation. The two boilers differed only in the firebox or furnace construction. The boiler designated as No. 1 has a furnace similar to the well-known *Hawley* down-draft type. The upper grate consists of heavy seamless steel water-tubes built into the furnace from the front inside head-sheet to a header extending from one side-sheet to the other. The lower grate is the Century rocking pattern.

The boiler designated as No. 2 has only the Century rocking grates applied in the ordinary manner.

Boiler No. 1:

Kewanee Smokeless Firebox Boiler, Size No. 111

Diameter of shell	48 in.
Length of boiler, over all	13 ft. 10 in.
Size of firebox or furnace	42 in. × 72 in.
Number and size of fire-tubes	48—3 in. × 91 in.
Total heating surface	488 sq. ft.
Upper grate—water-tubes, area	42 in. × 45 in.—13.1 sq. ft.
Lower grate—rocking, area	42 in. × 48 in.—14.0 sq. ft.

Boiler No. 2:

Kewanee Firebox Boiler, Size No. 12

Diameter of shell	48 in.
Length of boiler, over all	13 ft. 6 in.
Size of firebox or furnace	42 in. × 56 in.
Number and size of fire-tubes	48—3 in. × 103 in.
Total heating surface	502 sq. ft.
Grate—rocking, area	42 in. × 56 in.—16.3 sq. ft.

Coal.

The coals used for the tests were, Iowa slack, Illinois lump, Illinois washed nut, Illinois run of mine, and Pocahontas.

Method of Conducting Tests. Tests were conducted by making a careful estimate of the coal on the grates at the beginning of the test, and stopping the test when the fire was in the same condition as at the start.

Water was measured in carefully calibrated barrels, and pumped directly to the boilers. On account of the size of the pump it was impossible to have a steady flow of feed-water. During tests Nos. 1, 2, 3, 7, 8, and 9, feed-water was at a low temperature direct from source of supply. During tests Nos. 4, 5, 6, 10, 11, and 12, the feed-water was passed through a heater. At no time was the condensation returned to the boilers.

Steam pressure, temperature of feed-water, air and the chimney gases were taken at intervals of fifteen minutes.

The temperature of the gases, after passing through the boilers and before being returned around the outside of the boiler, was also observed.

Smoke. Smoke observations were taken on both boilers during five or six hours of each test, the standard of comparison being according to the *Ringelmann* chart. In all tests the length of the smoke observations covered sufficient time to include a complete cycle of operation, so that the results would have been the same if observations covering twenty-four hours had been made.

The percentage of density reported may be understood to be the per cent of time that black smoke would be produced to equal the smoke actually observed.

Efficiency. By efficiency is meant the percentage of total heat units in the coal that is absorbed by the boiler. These tests of both types of Kewanee Firebox Boilers show efficiencies as follows:

COMPARATIVE EFFICIENCIES OF SMOKELESS AND REGULAR FIREBOX BOILERS.

Kind of Coal	Smokeless Boiler	Regular Boiler	Per Cent Increase Smokeless Over Regular
Iowa slack	52.5	41.4	27
Illinois lump	74.3	65.2	14
Illinois washed nut	67.0	56.1	19
Illinois run of mine	59.3	48.5	22
Pocahontas	74.9	57.2	31

Boiler No. 1. (Size No. 111 Kewanee Smokeless Firebox Boiler.)

Test number	1	2	3	4	5	6
Duration of Trial.....Hrs.	10.00	10.10	9.37	11.95	10.10	9.31
Pressures:						
Barometric.....Ins.	29.45	29.25	28.81	28.98	29.07	29.29
Steam (Corrected Gage).....Lb.	5.15	5.40	4.32	3.10	4.42	5.00
Draft.....Ins.	0.24	0.23	0.18	0.21	0.24	0.22
Temperatures:						
Boiler Room.....Deg. F.	73.00	73.50	79.20	79.40	63.00	72.20
Flue Gases—Stack.....Deg. F.	337.00	296.00	256.00	286.00	280.00	343.00
Gases—Rear of Setting.....Deg. F.	......	......	588.00	591.00	636.00	637.00
Feed-Water.....Deg. F.	70.00	66.70	72.70	174.00	175.00	175.00
Kind of Fuel:	Iowa Slack	Illinois Lump	Illinois Nut	Iowa Slack	Illinois Mine Run	Pocahontas
Proximate Analysis:						
Moisture.....Per cent	11.93	7.84	4.62	8.42	5.26	0.54
Volatile Matter.....Per cent	28.06	34.08	32.00	30.80	32.40	15.26
Fixed Carbon.....Per cent	28.60	42.84	52.02	32.02	48.12	74.36
Ash.....Per cent	31.41	15.24	11.36	28.76	14.22	9.84
B.t.u. per lb. as fired.....	7691.00	10944.00	12190.00	8806.00	11609.00	14153.00
Total Fuel.....Lb.	2128.00	1561.50	1236.00	2546.00	1538.00	964.00
Dry Coal.....Lb.	1874.00	1439.00	1179.00	2319.00	1457.00	958.80
Total Ash.....Lb.	587.00	214.00	149.50	786.00	241.00	81.00
Combustible in Ash.....Per cent	23.01	31.65	28.70	27.00	15.67	48.23
Fuel per Square Foot Upper Grate per Hour:						
As Fired.....Lb.	16.25	11.80	16.08	16.26	11.62	7.90
Dry.....Lb.	14.30	10.87	9.60	14.80	11.02	7.86
Combustible.....Lb.	9.82	9.35	8.32	9.79	9.19	7.20
Evaporation:						
Feed-Water						
Per lb. Coal as Fired.....Lb.	3.98	7.27	7.63	4.52	7.15	10.80
Per lb. Dry Coal.....Lb.	4.52	7.88	8.00	4.96	7.55	10.99
Apparent Dry Steam:						
Per lb. Coal as Fired.....Lb.	......	......	7.36	4.08	6.81	10.48
Per lb. Dry Coal.....Lb.	......	......	7.71	4.48	7.14	10.52
Apparent Dry Steam from and at 212 Degrees:						
Per lb. Coal as Fired.....Lb.	4.58	8.39	8.43	4.27	7.10	10.92
Per lb. Dry Coal.....Lb.	5.20	9.09	8.83	4.69	7.49	10.98
Per Sq. Ft. Heating Surface.....Lb.	2.00	2.65	2.28	1.86	2.22	2.32
Efficiency Apparent:						
Boiler and Grate over all..Per cent	57.90	74.30	67.00	47.10	59.30	74.90
Gas Analysis:						
CO_2.....Per cent	4.90	4.30	6.42	4.90	5.08	5.26

COMPARATIVE EFFICIENCIES OF SMOKELESS AND REGULAR FIREBOX BOILERS.—*Continued.*

Boiler No. 1. (Size No. 111 Kewanee Smokeless Firebox Boiler.)

Test number	1	2	3	4	5	6
Density of Smoke—Per Cent of Time:						
No. 0 Smoke	63.40	39.40	52.40	27.50	44.10	86.00
No. 1 Smoke	27.10	45.40	22.90	62.50	28.60	13.20
No. 2 Smoke	8.70	13.30	18.70	9.80	20.70	0.00
No. 3 Smoke	0.70	1.80	5.70	0.20	6.30	0.80
No. 4 Smoke	0.10	0.10	0.30	0.00	0.30	0.00
No. 5 Smoke	0.00	0.00	0.00	0.00	0.00	0.00
Average Density.........Per cent	0.05	0.15	0.21	0.16	0.27	0.03

Boiler No. 2. (Size No. 12 Kewanee Regular Firebox Boiler.)

Test Number	7	8	9	10	11	12
Duration of Trial...........Hrs.	10.10	9.83	9.72	11.37	10.05	9.82
Pressures:						
Barometric..................Ins.	29.45	29.25	28.81	28.98	29.07	29.29
Steam (Corrected Gage).......Lb.	6.60	7.57	5.73	3.16	3.54	3.85
Draft.......................Ins.	0.20	0.21	0.20	0.22	0.24	0.23
Temperatures:						
Boiler Room.............Deg. F.	73.00	73.50	79.20	79.40	63.00	72.20
Flue Gases—Stack........Deg. F.	285.00	273.00	311.00	283.00	303.00	307.00
Gases—Rear Setting......Deg. F.	......	......	628.00	637.00	631.00	597.00
Feed-Water..............Deg. F.	70.10	67.20	72.30	171.00	174.00	176.00
Kind of Fuel:	Iowa Slack	Illinois Lump	Illinois Nut	Iowa Slack	Illinois Mine Run	Poca-hontas
Proximate Analysis:						
Moisture...............Per cent	11.93	7.84	4.62	8.42	5.26	0.54
Volatile Matter..........Per cent	28.06	34.08	32.00	30.80	32.40	15.26
Fixed Carbon...........Per cent	28.60	42.84	52.02	32.02	48.12	74.36
Ash....................Per cent	31.41	15.24	11.36	28.76	14.22	9.84
B.t.u. per lb. as Fired...........	7691.00	10944.00	12190.00	8806.00	11609.00	14153.00
Total Fuel..................Lb.	1693.00	1317.00	1188.50	2030.50	1295.00	846.00
Dry Coal....................Lb.	1491.00	1214.00	1133.70	1859.50	1226.80	841.40
Total Ash...................Lb.	566.00	179.00	196.50	619.00	228.50	68.00
Total Ash..............Per cent	33.40	13.60	16.53	30.50	17.65	8.04
Combustible in Ash......Per cent	33.00	33.00	36.50	18.14	25.06	32.70
Fuel per Square Foot Grate per Hour:						
As Fired....................Lb.	10.28	8.22	7.51	10.95	7.90	5.28
Dry.........................Lb.	9.06	7.58	7.16	10.03	7.52	5.26
Combustible.................Lb.	5.62	6.46	5.91	6.69	6.12	4.83
Evaporation:						
Feed-Water						
Per lb. Coal as Fired..........Lb.	3.04	6.37	6.55	4.40	6.69	9.61
Per lb. Dry Coal.............Lb.	3.46	6.90	6.87	4.81	7.08	9.66
Apparent Dry Steam:						
Per lb. Coal as Fired..........Lb.	......	......	5.48	3.44	5.56	8.03
Per lb. Dry Coal.............Lb.	......	......	5.74	3.75	5.88	8.08
Apparent Dry Steam from and at 212 Degrees:						
Per lb. Coal as Fired..........Lb.	3.51	7.36	6.31	3.59	5.80	8.34
Per lb. Dry Coal.............Lb.	3.99	7.97	6.60	3.93	6.12	8.39
Per Sq. Ft. Heating Surface....Lb.	1.17	1.96	1.53	1.28	1.49	1.43
Efficiency Apparent:						
Boiler and Grat oever all..Per cent	43.30	65.20	56.10	39.50	48.50	57.20

COMPARATIVE EFFICIENCIES OF SMOKELESS AND REGULAR FIREBOX BOILERS.—*Continued.*

Boiler No. 2. (Size No. 12 Kewanee Regular Firebox Boiler.)

Test number	7	8	9	10	11	12
Gas Analysis:						
CO_2 . . . . Per cent	3.00	5.50	4.50	5.22	4.37	4.40
Density of Smoke—Per Cent of Time:						
No. 0 Smoke	2.10	0.50	8.80	5.40	0.00	76.00
No. 1 Smoke	51.30	54.80	54.10	40.70	44.00	16.40
No. 2 Smoke	36.30	33.40	29.40	33.10	38.50	6.10
No. 3 Smoke	9.30	8.40	5.40	14.80	14.20	0.50
No. 4 Smoke	0.80	2.60	2.30	1.00	3.30	0.00
No. 5 Smoke	0.20	0.30	0.00	0.00	0.00	0.00
Average Density . . . . Per cent	0.31	0.32	0.27	0.33	0.35	0.06

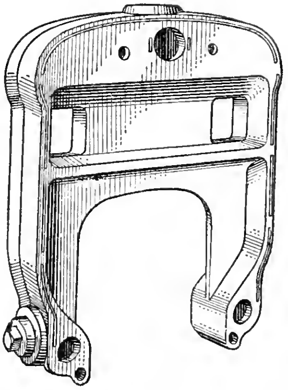


FIG. 2. CAST-IRON BOILER SECTION.

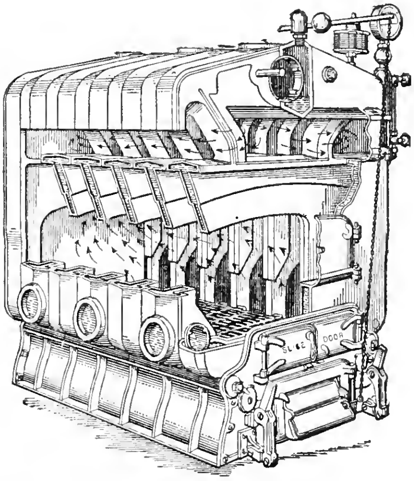


FIG. 3. NO. S-36-7 IDEAL STEAM BOILER.

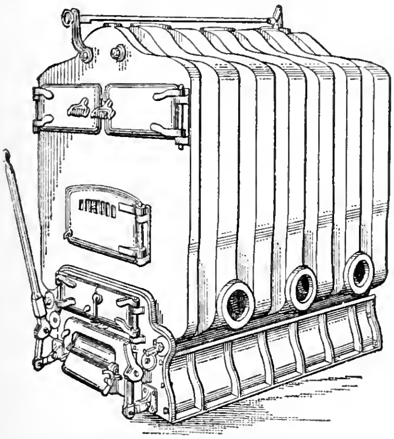


FIG. 4. NO. W-36-7 IDEAL WATER HEATER.

(American Radiator Co.)

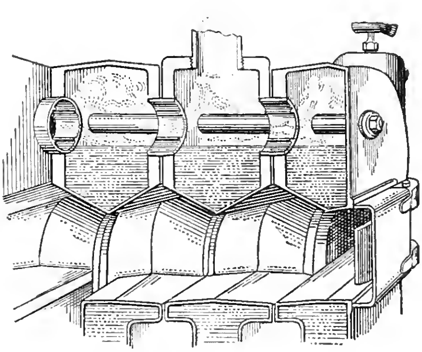


FIG. 5. CONNECTIONS OF THE HOLLOW CASTINGS IN IDEAL SECTIONAL BOILERS BY MACHINE-TURNED PUSH NIPPLES.

CAST-IRON HEATING BOILERS

Classification of Heating Boilers. Heating boilers are divided generally into *two classes—Cast Iron* and *Steel*—depending on the material of construction.

The cast-iron boilers are built up of separate sections, which are assembled on a suitable base and bolted together to form a unit (Fig. 2) of the desired heating and grate surface. The

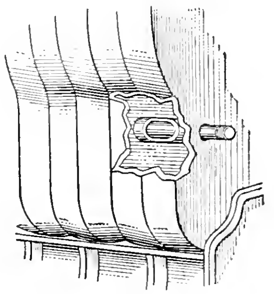


FIG. 6. CORED PASSAGEWAY FOR BINDING ROD.

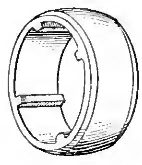


FIG. 7. CAST-IRON LATHE - TURNED PUSH NIPPLE.

sections are either arranged *vertically* and stand on end forming a *Sectional Boiler* (Figs. 3 and 4), or else are placed *horizontally* forming a *Round Boiler* (Figs. 11 and 12). The former arrangement is used almost exclusively for boilers of the larger sizes having a steam rating of 1800 sq. ft. and above.

Boilers *without headers* usually have two or more steam or water outlets, and for each section with one outlet two return inlets are provided, and all three openings are tapped and threaded the same size.

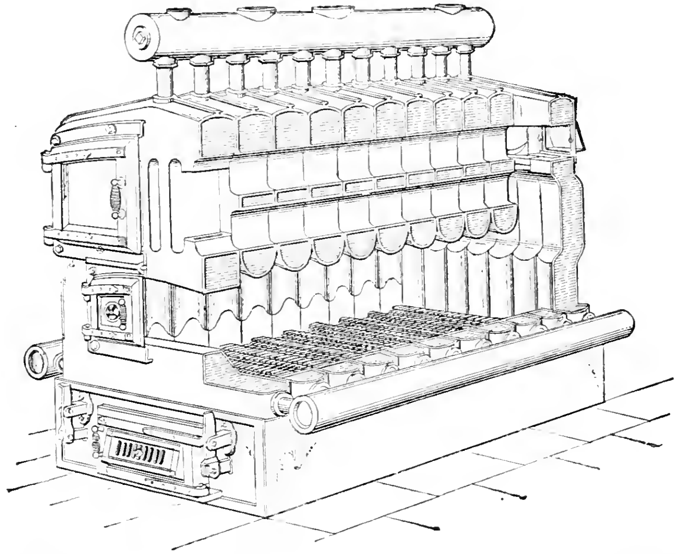


FIG. 8. MERCER CAST-IRON SECTIONAL BOILER, HEADER TYPE. (*H. B. Smith Co.*)

TABLE 4

DIMENSIONS AND CAPACITIES OF SECTIONAL CAST-IRON BOILERS

American Radiator Co.

Steam Boilers							Water Boilers*
Number	Length, Total Inches	Grate Area, Sq. Ft.	Ave. Area Firepot, Sq. Ft.	†Outlets, Ins.	Ash-Pit (Inside) Inches	8 Hour Rating Sq. Ft.	8 Hour Rating Sq. Ft.
S-15-4	40 7/8	1.95	2.47	2-3	20 15/16 x 21 5/8	300	500
S-15-5	47 1/8	2.60	3.30	2-3	20 15/16 x 27 13/16	425	700
S-15-6	53 3/8	3.25	4.10	2-3	20 15/16 x 34	550	900
Water Line, 40 7/8 in. Height, 53 1/2 in. Width, 34 1/2 in.‡							
S-19-5	51 3/4	3.32	4.00	2-3	20 x 29 15/16	600	1000
S-19-6	58 3/8	4.15	5.00	2-3	20 x 36 5/8	750	1250
S-19-7	65	4.98	6.00	3-3	20 x 43 5/16	900	1500
Water Line, 43 3/8 in. Height, 55 3/4 in. Width, 38 in.‡							
S-22-5	53 1/4	4.08	4.84	2-4	23 1/8 x 31 13/16	800	1300
S-22-6	60 1/4	5.10	6.05	2-4	23 1/8 x 38 7/8	1000	1650
S-22-7	67 1/4	6.12	7.26	3-4	23 1/8 x 45 15/16	1200	2000
Water Line, 46 1/4 in. Height, 59 1/2 in. Width, 42 in.‡							
S-25-5	59 1/4	5.44	6.48	2-4	28 x 35 3/16	1100	1825
S-25-6	66 7/8	6.80	8.10	2-4	28 x 42 7/8	1350	2225
S-25-7	74 1/2	8.16	9.72	3-4	28 x 50 9/16	1600	2650
S-25-8	82 1/4	9.52	11.34	3-4	28 x 58 1/4	1850	3050
Water Line, 51 in. Height, 64 1/8 in. Width, 47 1/4 in.‡							
S-28-5	60	6.24	7.33	2-4	30 5/8 x 35 1/2	1300	2150
S-28-6	68	7.80	9.16	2-4	30 5/8 x 43 1/2	1625	2675
S-28-7	76	9.36	10.99	3-4	30 5/8 x 51 1/2	1950	3200
S-28-8	84	10.92	12.83	3-4	30 5/8 x 59 1/2	2275	3725
Water Line, 53 3/8 in. Height, 67 1/16 in. Width, 50 1/2 in.‡							
S-36-5	69 3/4	9.12	10.40	2-5	38 15/16 x 40 3/4	2100	3450
S-36-6	78 7/8	11.40	13.00	2-5	38 15/16 x 49 7/8	2625	4325
S-36-7	88	13.68	15.60	3-5	38 15/16 x 59	3150	5200
S-36-8	97 1/8	15.96	18.20	3-5	38 15/16 x 68 1/8	3675	6050
S-36-9	106 1/4	18.24	20.80	4-5	38 15/16 x 77 1/4	4200	6925
§Water Line, 60 3/4 in. Height, 76 1/4 in. Width, 60 in.‡							
S-48-6	92	18.00	18.75	2-6	52 x 54 1/2	5275	8700
S-48-7	102 3/4	21.60	22.50	2-6	52 x 65 1/4	6300	10375
S-48-8	113 1/2	25.20	26.25	3-6	52 x 76	7325	12050
S-48-9	124 1/4	28.80	30.00	3-6	52 x 86 3/4	8350	13725
S-48-10	135	32.40	33.75	3-6	52 x 97 1/2	9375	15400
Water Line, 72 in. Height, 97 in. Width, 80 in.‡							

* Water boilers are always indicated by prefix W as W-15-4.
† For each outlet there are two inlets of same size—one on each side.
‡ Heights and Widths apply to hoth steam and water boilers, and are inclusive of trimmings.
§ 48-Inch steam boilers are furnished with four 4-inch and 48-inch water boilers with four 6-inch return tappings, two on the face of back section, and one on each side of boiler in third section from rear. Back openings should be yoked together so that both halves of boiler may be drained equally.
For smoke pipe and other measurements, see Table 5.
All dimensions here given are same for steam and water boilers.

Boilers *with headers* (Fig. 8) are made up of independent sections, each of which has one outlet and two inlets connecting to the flow and return headers respectively. There is no connection between the individual sections in this case.

Sectional Cast-Iron Boilers. Sectional cast-iron boilers may be assembled with cast-iron *slip* or *push nipples* (Fig. 7) between the *sections* (Fig. 2), which are held together by binding rods at top and bottom (Figs. 5 and 6), there usually being three such nipples for each section, or else the sections may be connected to steam and return headers with no direct connections between the sections (Fig. 8) using *screw nipples*. The push nipples are machine-turned and accurately tapered to the 1/1000 of an inch, making a metal to metal joint.

It will be seen by reference to Figs. 3 and 4 that the steam boiler sections are so shaped as to give somewhat greater volume above the water line than exists in the upper part of the water heaters or "boilers." It is, however, possible to use a steam boiler for water heating, although the reverse is not so true.

Cast-Iron Boiler Data. *Catalog ratings* should always be checked, as already explained, if same are to be used in selecting a boiler to supply a given amount of radiation, and due allowance made for radiation losses from connecting piping, such as mains, branches and risers. See "Rating of Heating Boilers" for method of checking and determining size of boiler for actual conditions.

Dimension drawings are shown of typical modern sectional cast-iron steam and water boilers (Figs. 9 and 10), as well as of round cast-iron steam boilers and water heaters (Fig. 13).

Dimension tables for Figs. 9, 10 and 13, for all sizes, and tables of the more important data concerning each size of boiler have been prepared, and so consolidated as to facilitate reference in selecting a boiler or laying out a heating system.

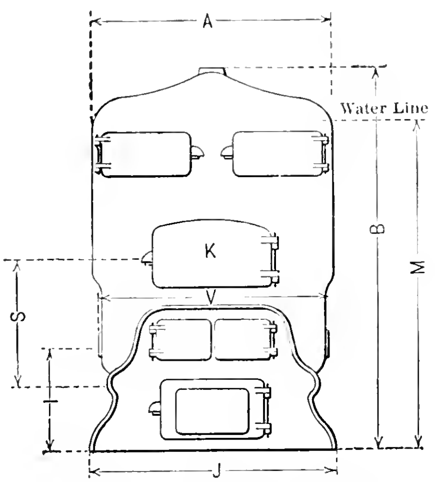


FIG. 9. Front View. See Table 5.

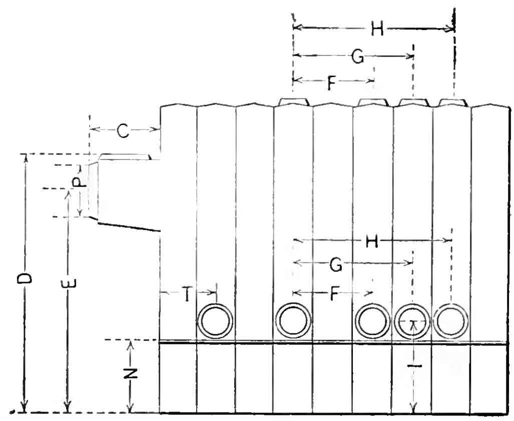


FIG. 10. Side View. For details of measurement see Table 5. Boilers are so designed that any casting, whether round or square, may be taken through any door or opening which is not less than 2 feet 6 inches wide.

TABLE 5

IDEAL SECTIONAL BOILER MEASUREMENTS

American Radiator Co.

Distances in inches on the outlines shown in Figs. 9 and 10

	15-In. Boilers		19-In. Boilers		22-In. Boilers	
	Water	Steam	Water	Steam	Water	Steam
A	27 1/2	28 1/2	31 1/4	32 1/4	35 1/4	36 1/4
B	42 7/16	46 3/16	50	50	53	53
*C	13 5/16	13 5/16	15 5/8	15 5/8	15 1/4	15 1/4
†D	41 1/8	41 1/8	45 1/8	45 1/8	47 3/4	47 3/4
E	34 3/4	34 3/4	37 3/4	37 3/4	40 1/2	40 1/2
F	12 1/2	12 1/2	13 1/4	13 1/4	14 1/8	14 1/8
G	18 3/4	18 3/4	19 7/8	19 7/8	21 1/4	21 1/4
H	25	25	26 1/2	26 1/2	28 1/4	28 1/4
I	16 3/16	16 3/16	16	16	16 3/4	16 3/4
J	23 3/4	23 3/4	26	26	29 1/8	29 1/8
K	8x14	8x14	‡8x14	‡8x14	‡8x14	‡8x14
M	....	40 7/8	....	43 3/8	....	46 1/4
N	11 3/4	11 3/4	9 3/8	9 3/8	9 1/2	9 1/2
P	8	8	9	9	10	10
S	13 1/4	13 1/4	12 5/8	12 5/8	12 3/8	12 3/8
T	7 1/2	7 1/2	8	8	8 1/2	8 1/2
V	25 13/32	25 13/32	29 5/8	29 5/8	33 9/16	33 9/16

TABLE 5.—*Continued.*

	25-In. Boilers		28-In. Boilers		36-In. Boilers		48-In. Boilers	
	Water	Steam	Water	Steam	Water	Steam	Water	Steam
A	40 3/8	41 3/8	43 1/2	44 1/2	53 1/4	54 1/4	68	69
B	57 7/8	57 7/8	60 5/8	60 5/8	69 1/8	69 1/8	81 3/4	81 3/4
*C	17 1/2	17 1/2	18 1/8	18 1/8	21 11/16	21 11/16	27 1/4	27 1/4
†D	53	53	55 7/8	55 7/8	63 3/8	63 3/8	73 1/8	73 1/8
E	44 1/8	44 1/8	46 1/4	46 1/4	52 9/16	52 9/16	59 1/2	59 1/2
F	15 3/8	15 3/8	16	16	18 1/4	18 1/4	21 1/2	21 1/2
G	23 1/16	23 1/16	24	24	27 3/8	27 3/8	32 1/4	32 1/4
H	30 3/4	30 3/4	32	32	36 1/2	36 1/2	43	43
I	17 3/4	17 3/4	17 7/8	17 7/8	18 7/16	18 7/16	22 3/8	22 3/8
J	34 1/2	34 1/2	37 1/8	37 1/8	45 7/16	45 7/16	58 3/8	58 3/8
K	‡9x18	‡9x18	‡9x18	‡9x18	‡10x20	‡10x20	11x19	11x19
M	....	51	....	53 3/8	....	60 3/4	....	72
N	9 7/8	9 7/8	10	10	10 13/16	10 13/16	14 11/16	14 11/16
P	11	11	12	12	15	15	21	21
S	14 1/4	14 1/4	14 1/4	14 1/4	15 5/8	15 5/8	17 3/4	17 3/4
T	9 1/8	9 1/8	9 1/2	9 1/2	10 7/8	10 7/8	12 3/4	....
V	39 3/8	39 3/8	41 13/16	41 13/16	52 5/8	52 5/8	64 11/16	64 11/16

* Measured without smoke-hood cover.
† Measured with smoke-hood cover on.
‡ For wood, feed door K in 19-inch boilers is 10 1/4 x 18 inches; in 22-inch boilers is 11 1/8 x 18 inches; in 25-inch boilers is 11 1/8 x 18 inches; in 28-inch boilers, is 12 13/16 x 19 7/8 inches; in 36-inch boilers, is 13 15/16 x 24 inches.
Do not bush the flow-pipe outlets of steam boilers; connect all of them full size to the main.

Round Cast-Iron Boilers. The round boilers are built up of *horizontal* sections in a manner similar to that used for the sectional boilers already described.

The *capacity and dimension data* for these boilers will be found in Tables 6, 7, 8, and 9 and Fig. 13.

Smokeless or Down-Draft Cast-Iron Boilers. Boilers having a water grate are now being made for use with free burning soft coal, where local smoke ordinances would not permit the use of such fuel on ordinary grates.

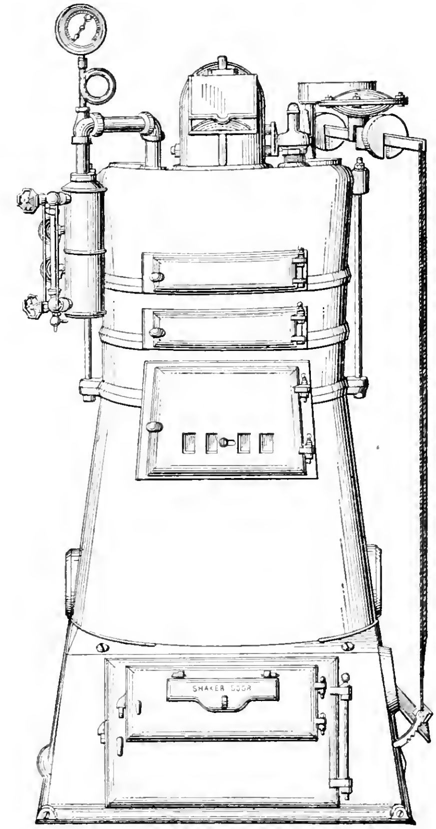
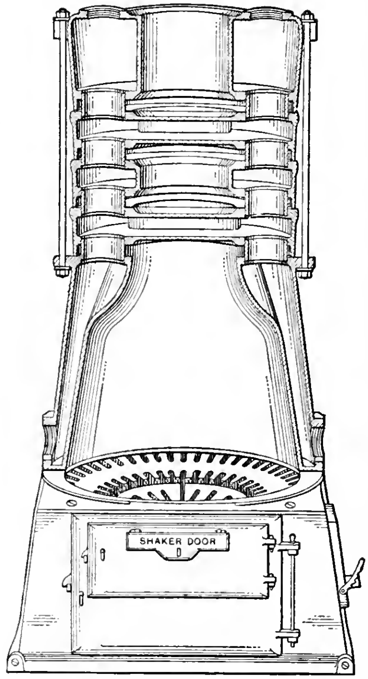


FIG. 11. CAPITOL-WINCHESTER STEAM BOILER.



Sectional View.

FIG. 11a. CAPITOL-WINCHESTER STEAM BOILER.

TABLE 6

CAPITOL-WINCHESTER STEAM BOILERS—CAPACITIES

United States Radiator Corporation

No.	Rating Sq. Feet	Actual Grate Diam., Inches	Grate Area, Sq. Feet	Height Water Line, Inches	Height Outlets, Inches	Outlets and Inlets, Inches	Smoke Pipe, Inches
3130	200	15	1.23	44 3/16	49 3/16	2-2 1/2	6
3140	225	15	1.23	48 3/16	53 9/16	2-2 1/2	6
3230	250	17	1.58	44 3/8	49 1/2	2-2 1/2	6
3240	300	17	1.58	49	54 1/16	2-2 1/2	6
3330	325	20	2.18	44 3/16	49 15/16	2-2 1/2	7
3340	375	20	2.18	49	54 3/4	2-2 1/2	7
3350	425	20	2.18	53 13/16	59 9/16	2-2 1/2	7
3440	500	24 1/2	3.27	50 5/8	56 1/8	2-3	8
3450	575	24 1/2	3.27	55 1/2	61	2-3	8
*3460	650	24 1/2	3.27	60 5/16	65 13/16	2-3	8
3540	750	29	4.59	52 1/16	57 9/16	2-4	9
3550	850	29	4.59	56 15/16	62 7/16	2-4	9
*3560	950	29	4.59	61 13/16	67 5/16	2-4	9
3640	1100	33	5.94	53 9/16	59 1/16	2-4	10
3650	1225	33	5.94	58 7/16	63 15/16	2-4	10
*3660	1350	33	5.94	63 5/16	68 13/16	2-4	10

Detailed dimensions, Table 8.
Boiler trimmings extend 16 inches above outlets.
Use the next size larger fire pot for soft coal.

* Strong draft is necessary when these boilers are used with soft coal.
3600 series boilers are equipped with triangular grates.

TABLE 7

CAPITOL-WINCHESTER WATER BOILERS—CAPACITIES

United States Radiator Corporation

No.	Rating Sq. Feet	Actual Grate Diam., Inches	Grate Area, Sq. Feet	Height Outlets, Inches	Outlets and Inlets, Inches	Smoke Pipe, Inches
4130	325	15	1.23	43 15/16	2–2 1/2	6
4140	375	15	1.23	47 15/16	2–2 1/2	6
4230	425	17	1.58	44 1/4	2–2 1/2	6
4240	500	17	1.58	48 13/16	2–2 1/2	6
4330	550	20	2.18	44 11/16	2–2 1/2	7
4340	625	20	2.18	49 1/2	2–2 1/2	7
4350	700	20	2.18	54 5/16	2–2 1/2	7
4440	825	24 1/2	3.27	50 7/8	2–3	8
4450	950	24 1/2	3.27	55 3/4	2–3	8
*4460	1075	24 1/2	3.27	60 9/16	2–3	8
4540	1225	29	4.59	52 5/16	2–4	9
4550	1400	29	4.59	57 3/16	2–4	9
*4560	1575	29	4.59	62 1/16	2–4	9
4640	1825	33	5.94	53 13/16	2–4	10
4650	2025	33	5.94	58 11/16	2–4	10
*4660	2225	33	5.94	63 9/16	2–4	10

Detailed dimensions, Table 9.
Use the next size larger fire pot for soft coal.
*Strong draft is necessary when these boilers are used with soft coal.
3600 series boilers are equipped with triangular grates.

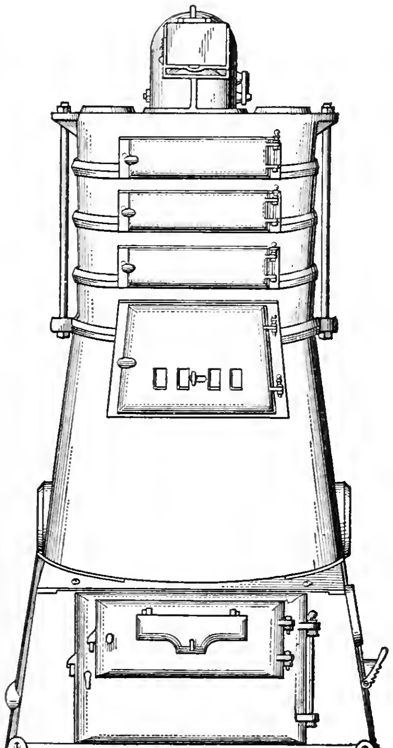


FIG. 12. CAPITOL-WINCHESTER WATER BOILER.

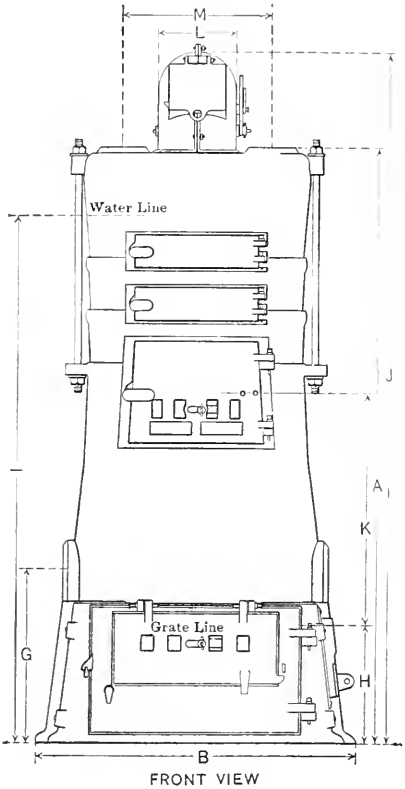


FIG. 13. For details of measurement see Tables 8 and 9.

TABLE 8

CAPITOL-WINCHESTER STEAM BOILERS—MEASUREMENTS

United States Radiator Corporation

Size	A	B	C	G	H	I	J	K	L	M
3130	49 3/16	24 1/4	8 x 8	16 1/8	8 7/8	44 3/16	56 3/8	23 1/2	6	13 11/16
3140	53 9/16	24 1/4	8 x 8	16 1/8	8 7/8	48 3/16	60 3/4	23 1/2	6	13 11/16
3230	49 1/2	26 1/4	8 x 9	16 1/8	8 7/8	44 3/8	56 11/16	23 1/2	6	13 11/16
3240	54 1/16	26 1/4	8 x 9	16 1/8	8 7/8	49	61 1/4	23 1/2	6	13 11/16
3330	49 15/16	29 5/16	9 x 11	16 1/8	8 7/8	44 3/16	58 1/8	23 1/2	7	13 11/16
3340	54 3/4	29 5/16	9 x 11	16 1/8	8 7/8	49	62 15/16	23 1/2	7	13 11/16
3350	59 9/16	29 5/16	9 x 11	16 1/8	8 7/8	53 13/16	67 3/4	23 1/2	7	13 11/16
3440	56 1/8	35	9 x 12	17 1/2	9 7/8	50 5/8	65 5/16	24 1/8	8	16 5/16
3450	61	35	9 x 12	17 1/2	9 7/8	55 1/2	70 3/16	24 1/8	8	16 5/16
3460	65 13/16	35	9 x 12	17 1/2	9 7/8	60 5/16	75	24 1/8	8	16 5/16
3540	57 9/16	40	9 x 13	19	10 9/16	52 1/16	67 11/16	24 11/16	9	17 13/16
3550	62 7/16	40	9 x 13	19	10 9/16	56 15/16	72 9/16	24 11/16	9	17 13/16
3560	67 5/16	40	9 x 13	19	10 9/16	61 13/16	77 7/16	24 11/16	9	17 13/16
3640	59 1/16	44 3/4	9 x 14	20 1/2	12 1/16	53 9/16	70 3/16	24 11/16	10	21 7/16
3650	63 15/16	44 3/4	9 x 14	20 1/2	12 1/16	58 7/16	75 1/16	24 11/16	10	21 7/16
3660	68 13/16	44 3/4	9 x 14	20 1/2	12 1/16	63 5/16	79 15/16	24 11/16	10	21 7/16

TABLE 9

CAPITOL-WINCHESTER WATER BOILERS—MEASUREMENTS

United States Radiator Corporation

Size	A	B	C	G	H	J	K	L	M
4130	43 15/16	24 1/4	8 x 8	16 1/8	8 7/8	51 1/8	23 1/2	6	13 11/16
4140	47 15/16	24 1/4	8 x 8	16 1/8	8 7/8	55 1/2	23 1/2	6	13 11/16
4230	44 1/4	26 1/4	8 x 9	16 1/8	8 7/8	51 7/16	23 1/2	6	13 11/16
4240	48 13/16	26 1/4	8 x 9	16 1/8	8 7/8	56	23 1/2	6	13 11/16
4330	44 11/16	29 5/16	9 x 11	16 1/8	8 7/8	52 7/8	23 1/2	7	13 11/16
4340	49 1/2	29 5/16	9 x 11	16 1/8	8 7/8	57 11/16	23 1/2	7	13 11/16
4350	54 5/16	29 5/16	9 x 11	16 1/8	8 7/8	62 1/2	23 1/2	7	13 11/16
4440	50 7/8	35	9 x 12	17 1/2	9 7/8	60 1/16	24 1/8	8	16 5/16
4450	55 3/4	35	9 x 12	17 1/2	9 7/8	64 15/16	24 1/8	8	16 5/16
4460	60 9/16	35	9 x 12	17 1/2	9 7/8	69 3/4	24 1/8	8	16 5/16
4540	52 5/16	40	9 x 13	19	10 9/16	62 7/16	24 11/16	9	17 13/16
4550	57 3/15	40	9 x 13	19	10 9/16	67 5/16	24 11/16	9	17 13/16
4560	62 1/16	40	9 x 13	19	10 9/16	72 3/16	24 11/16	9	17 13/16
4640	53 13/16	44 3/4	9 x 14	20 1/2	12 1/16	64 15/16	24 11/16	10	21 7/16
4650	58 11/16	44 3/4	9 x 14	20 1/2	12 1/16	69 13/16	24 11/16	10	21 7/16
4660	63 9/16	44 3/4	9 x 14	20 1/2	12 1/16	74 11/16	24 11/16	10	21 7/16

The *capacities* of the *American Radiator Co.'s* smokeless steam boilers range from 1500 to 2700 sq. ft. by increments of 300 sq. ft., and from 3000 to 6000 sq. ft. by increments of 500 sq. ft. The smaller series carry a 54 7/16 in. water line and the larger a 60 3/4 in. water line. The water heaters of this smokeless type range from 2450 sq. ft. capacity up to 9600 sq. ft. in two series, corresponding to the steam boilers. The increase in capacity per boiler up to 4450 sq. ft. is 500 sq. ft., and above 4800 sq. ft. capacity the increment is 800 sq. ft. per boiler.

Exceptionally good draft and also *perfect chimney and flue construction* are demanded whereever these boilers are installed. Minimum allowable heights and diameters or areas of chimney are specified and insisted upon by the manufacturers of these boilers.

Fuels for Cast-Iron Down-Draft Boilers. The ratings of down-draft boilers are based on the use of free-burning soft coal. Caking coals which fuse and check the air supply require more stoking, and, when such coals are used, the above ratings should be discounted at least 25 per cent. This applies to all down-draft boilers when caking coal is used.

These boilers will burn any kind of soft coal without generating smoke. It has been demonstrated by a large number of thermal tests that a free-burning coal, run of mine, gives the best results. Since the course of the gases is through the grates, it is obvious that any kind of

coal that will close or choke the air spaces between the grates will shut off the air necessary to combustion.

Caking coal will do this, because it fuses and forms a mass through which the gases pass slowly; hence the necessity usually arises for frequent stoking when caking coal is used in order to keep the gas passages open.

Selection of Cast-Iron Boilers. The *selection* of cast-iron boilers should not be influenced too largely by considerations of price, and the ease with which they may be carried into a building where structural conditions would interfere with the introduction of a steel boiler. In many cases the character of the service or attendance, or both, especially in government and other public building work, may be such that steel equipment, which is capable of withstanding more abuse, should be used.

If cast-iron boilers are to be installed, the grate area necessary should be carefully computed, as already indicated, using an average rate of combustion, and fuel pot depth based on the firing period required.

The *U. S. Treasury Dept.* selects cast-iron boilers by proportioning them to carry 25 per cent more radiation than actually installed if anthracite coal is used, and 35 per cent more if bituminous coal is used. In *addition to this,* suitable allowance must be made for mains and other piping, and in most cases *two boilers are installed,* each capable of supplying $\frac{2}{3}$ of the radiation in order to provide for units which can be operated with a high load factor, and also serve as a *reserve* for each other in case of break down. See also "Cast-Iron and Steel Heating Boilers Compared."

STEEL HEATING BOILERS

Boilers made of *wrought iron* or *mild steel* have long been in use for heating, and are generally classed as *portable,* that is, without brick setting or as *brick set,* the latter having an enclosure of brick forming the sides of the furnace and combustion chamber.

Portable Steel Boilers. The *portable type* may be either *vertical* or *horizontal;* the smaller sizes being made usually in the former, and the larger (Fig. 14) in the latter shape. Round vertical steel boilers of more than 1400 sq. ft. rated capacity of steam radiation are seldom used.

The large portable boilers of the horizontal type are either of the *return tubular,* the *firebox,* or the *internally fired* marine type as shown under "Power Boilers." In government work the Century made by *Kewanee Boiler Co.*, and the Ideal Cylindrical boiler made by *American Radiator Co.* have been used extensively, but the firebox type either with or without brick setting, such as the Kewanee Firebox (Figs. 14 and 17 to 19) is beginning to take the place of the former two.

Steel Boiler Data. The *capacities of the portable* or brick set steel boilers of the vertical type are about the same as for small cast-iron boilers of the same grate area, and range from 400 to 1400 sq. ft. of steam radiation.

The most common of these boilers are the *Dunning,* the *Gorton,* and the *Kewanee,* and reference to the makers' catalogs will give the detailed capacities, sizes, and dimensions in the same way as already shown and listed for cast-iron boilers.

The *capacities and dimensions* of the Kewanee Portable Firebox Boilers (Fig. 14) are given in Tables 10 to 13. These boilers may be obtained in either the direct-draft or down-draft smokeless type, but only the former are here shown, and the data tabulated.

This boiler is also built for brick setting (Fig. 18) similar to the setting shown for a smokeless boiler (Figs. 17 and 19). See manufacturer's catalog for data on direct-draft brick-set boiler, as data given apply to portable type only.

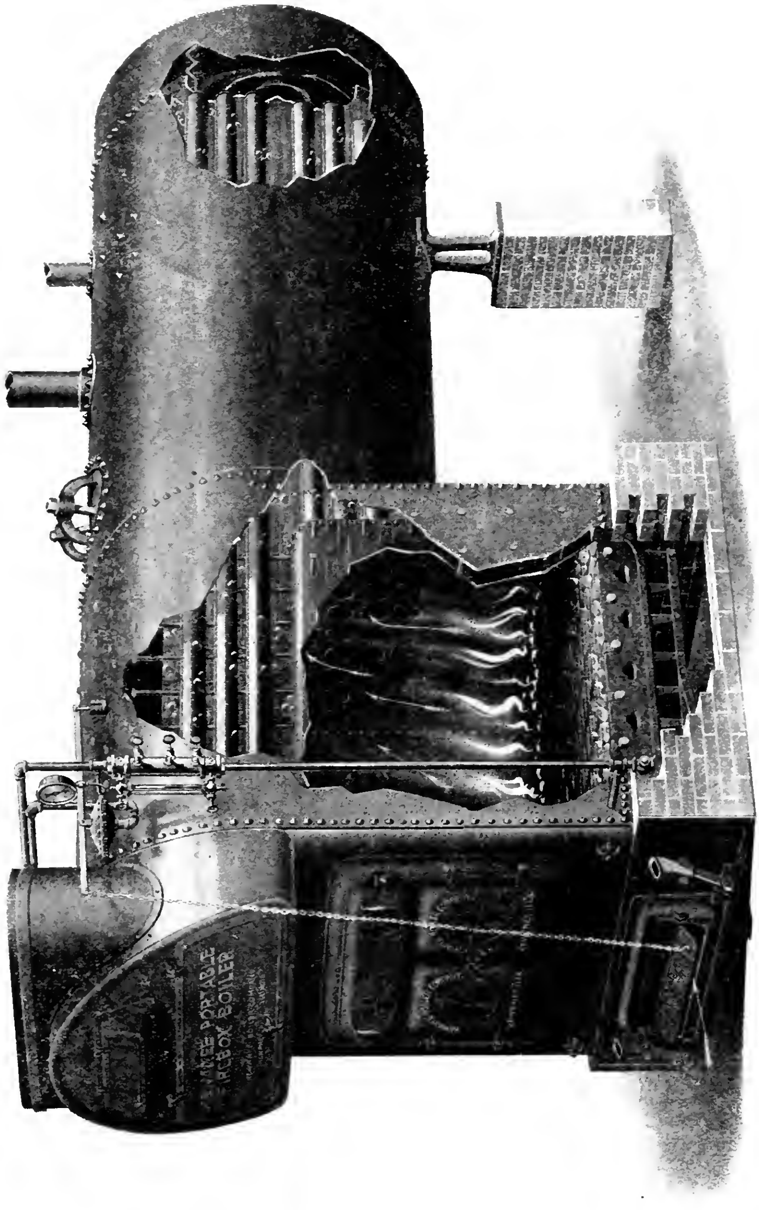


Fig. 14. Kewanee Firebox Boiler. Portable Type.

TABLE 10

CAPACITIES—KEWANEE FIREBOX BOILERS—PORTABLE TYPE

Number	0000	000	401	402	403	404	405	406	407	408	409
Capacity, Steam.......square feet	500	700	900	1100	1300	1500	1800	2100	2400	2800	3300
Capacity, Water... ...square feet	800	1100	1500	1800	2100	2500	2900	3400	4000	4600	5400
Approximate weight.......pounds	2900	3200	4100	4500	4900	5500	6000	6500	7600	8600	9100

CAPACITIES—KEWANEE FIREBOX BOILERS—PORTABLE TYPE

Number	410	411	412	413	414	415	416	417	418	419	420
Capacity, Steam....square feet	3800	4300	4800	5500	6000	7000	8000	9000	10000	12000	13000
Capacity, Water....square feet	6200	7000	7800	9000	9800	11400	13000	15000	17000	20000	22000
Approximate weight....pounds	10000	11000	12000	13000	14000	16000	17500	20000	22000	23000	24000

NOTE.—All spacing rings are semi-steel, and all boilers with Nos. 0000 to 409 inclusive have cast-iron base instead of brick as shown in Fig. 15.

Firebox Boiler Construction. The *firebox boilers* are usually built of mild, open hearth steel having a tensile strength of 60,000 lb. per sq. in. and are good up to 60 lb. working pressure, and are so insured by the boiler insurance companies. By making use of wrought iron space rings around the legs of the firebox, and by using extra stays and braces these boilers may be built for 100 lb. working steam pressure, and will be so insured. These boilers are assembled complete at the factory, and hence are handled as a complete unit during installation.

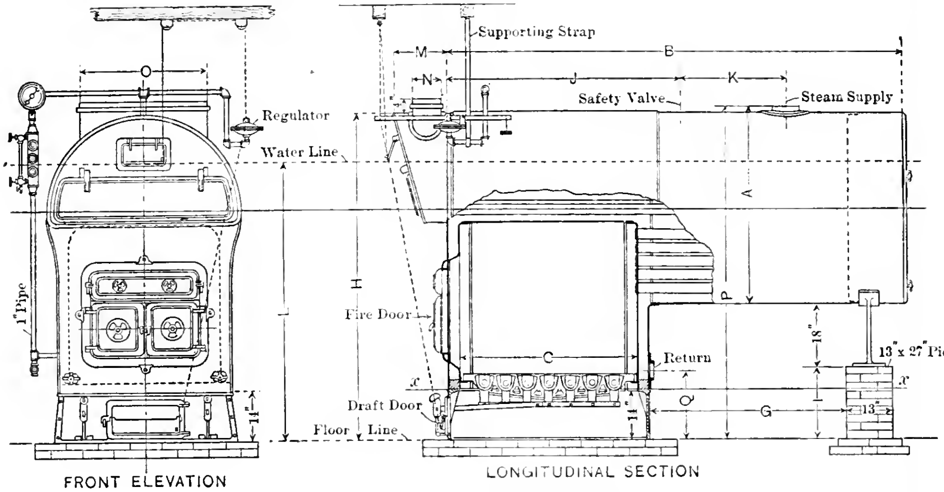
The grate and combustion chamber above it are surrounded by a water jacketed firebox (Figs. 14 and 17) so that a large amount of *direct heating surface* is exposed to the fire and hot gases, and the direct fire tubes leading out of the back of this chamber, together with the outside surface of the shell, below the water line, which, in the brick set type is enclosed by the brick setting, furnish the *indirect heating surface* to absorb the heat in the burned gases and reduce their temperature to within 25° to 50° F. of that of the steam in the boiler. The flue gases from these boilers are seldom above 300° F., and hence ample chimney capacity must be provided to give the necessary draft.

Brick Set Firebox Boilers. The brick set steel heating boiler is usually of the firebox type (Figs. 17, 18, or 19), or else of the horizontal return tubular type, and is preferably of the smokeless type for soft coal.

Smokeless Firebox Boilers. Smokeless boilers (Figs. 17 and 19) are being very generally used at the present time in order to overcome the smoke nuisance, due to the use of soft coal, as required by the smoke ordinances in force in many localities. These boilers are very similar to the ordinary firebox boiler except in the number of grates and the arrangement of the combustion chamber. An inclined *water grate* is built across the firebox or combustion chamber as shown in Fig. 17 and an inverted bridge wall compels the air, which is admitted above this grate, to pass down through the fuel placed on same and over the lower grate before entering the fire tubes. The *lower grate* is of the ordinary rocking type (Figs. 45 and 46), and the hot coal which accumulates on this grate serves to burn completely the volatile gases given off from the fuel above before they can be condensed into smoke.

These boilers are nothing more than a combination in a single unit of the down draft or *Hawley* type of furnace and the ordinary firebox boiler. When using soft coal, the makers claim these boilers utilize from 14 to 30 per cent more of the heat in the coal than do the ordinary boilers. Smokeless boilers may be either portable or brick set.

The *capacity and dimension data* for the Kewanee Smokeless Firebox Boiler "Brick set" are given in Table 14. These boilers are also built in portable form in a style similar to that shown in Figs. 15 and 16, but have water grates as shown in Fig. 17.



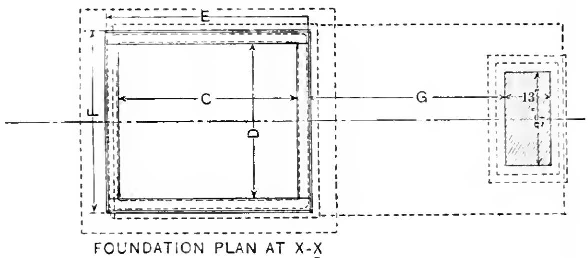
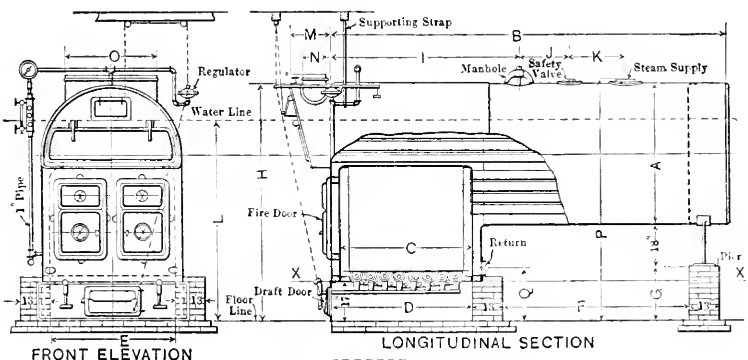


FIG. 15. SETTING PLAN FOR FIRE-BOX BOILER. PORTABLE TYPE. CAST-IRON ASH-PIT. SEE TABLE 11.



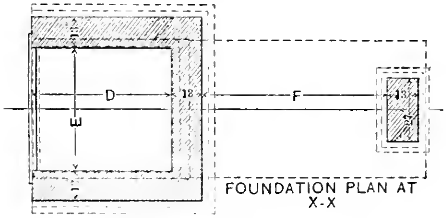


FIG. 16. SETTING AND FOUNDATION PLAN FOR FIREBOX BOILER. PORTABLE TYPE. BRICK ASH-PIT. (SEE TABLE 12.)

TABLE 11

SETTING MEASUREMENTS—PORTABLE FIREBOX BOILERS

Alternate sizes omitted—see Table 10

Number of Boiler	0000	401	403	405	407	409	410	411	412	413	414
Diameter of Boiler, inches . . . A	30	36	36	42	48	48	54	54	54	60	60
Length of Boiler, feet and inches . . . B	5-5 1/8	5-11 1/8	7-1/8	7-8 1/8	8-3 1/4	10-9 1/4	9-8 1/4	10-9 1/4	11-9 1/4	12-1 1/4	13-3 1/4
Length of Firebox, inches . . . C	20	26	38	38	38	50	44	50	56	56	62
Width of Firebox, inches . . . D	18 3/8	24 3/8	24 3/8	30 3/8	36 3/8	36 3/8	.....	.....	.....	.....	.....
Length Ash Pit Base, inches . . . E	27 1/2	33 1/2	45 1/2	45 1/2	45 1/2	57 1/2	.....	.....	.....	.....	.....
Width Ash Pit Base, inches . . . F	27 1/2	33 1/2	33 1/2	39 1/2	45 1/2	45 1/2	.....	.....	.....	.....	.....
Ash Pit Base to Pier, inches . . . G	27	27	28	36	39	57	.....	.....	.....	.....	.....
Total Height, inches . . . H	59 1/4	68 9/16	68 9/16	79 9/16	84 5/8	84 5/8	92 5/16	92 5/16	92 5/16	97 5/16	97 5/16
Height of Pier, inches . . . I	11	14 5/16	14 5/16	19 5/16	18 5/16	18 5/16	20	20	20	19 5/8	19 5/8
Location Safety Valve, feet and inches . . . J	2-10	3-4	4-4	4-5	4-6	5-6	5-4	5-6	8-1/2	7-11 1/4	8-8
Location Steam Supply, inches . . . K	12	11	11 1/8	15	17	30	24	30	18	16	24
Height of Water Line, inches . . . L	53 1/2	66 5/16	66 5/16	69 1/2	72 5/16	72 5/16	78 1/2	78 1/2	78 1/2	83	83
Length of Smoke Box, inches . . . M	8 3/4	11	11	12 3/8	14 1/2	14 1/2	14 3/4	14 3/4	14 3/4	16 3/4	16 3/4
Width Breeching Connection, inches . . . N	6	8	8	8	8	8	8	8	8	10	10
Length Breeching Connection, inches . . . O	14	18	18	22	28	28	36	36	36	42	42
Height Steam Supply, inches . . . P	59 3/4	70 1/16	70 1/16	81 1/16	86 1/4	86 1/4	93 11/16	93 11/16	93 11/16	99 1/8	99 1/8
Height Return, inches . . . Q	17 3/4	18 1/4	18 1/4	18 3/4	19 1/4	19 1/4	19 1/4	19 1/4	19 1/4	19 3/4	19 3/4
Number of Common Brick	150	210	230	260	290	330	650	700	750	800	850
Outside Surface to be Covered, sq. ft.	50	65	75	90	115	150	155	175	185	190	220
Length of Ash Pit, inches . . . D	.....	.....	.....	.....	.....	.....	47 1/2	53 1/2	59 1/2	59 1/2	65 1/2
Width of Ash Pit, inches . . . E	.....	.....	.....	.....	.....	.....	43	43	43	49	49
Ash Pit, wall to pier, ft. and in . . . F	.....	.....	.....	.....	.....	.....	3-9	4-4	4-10	5-2	5-10

Boilers Nos. 410, 411, 412, 413 and 414 are constructed as shown in Fig. 15, but are set on brick base instead of cast iron ash pit. For key letters D, E and F pertaining to these sizes see Fig. 16.

TABLE 12

SETTING AND FOUNDATION MEASUREMENTS—PORTABLE FIREBOX BOILERS

Number of Boiler	415	416	417	418	419	420
Diameter of Boiler, inches . . . A	60	60	66	66	72	72
Length of Boiler, feet and inches . . . B	13-11 7/8	15-9 7/8	15-6	16-11 7/8	15-7 7/8	17-1 7/8
Length of Grade, inches . . . C	56	62	62	68	68	74
Length of Ash Pit, inches . . . D	60	66	66	72	72	78
Width of Ash Pit, inches . . . E	54	54	60	60	66	66
Ash Pit Wall to Pier, feet and inches . . . F	6-7	7-11	7-5	8-5	6-11	7-11
Height of Pier, inches . . . G	23	23	23	23	23	23
Total Height, inches . . . H	101 5/16	101 5/16	107 3/8	107 3/8	113 3/8	113 3/8
Location of Manhole, feet and inches . . . I	6-8	7-3 1/2	7-1	7-6	7-8	8-3
Location of Safety Valve, inches . . . J	21	27	22	33	24	24
Location of Steam Supply, inches . . . K	24	30	43	42	35	43
Height of Water Line, inches . . . L	86	86	90	90	97	97
Length Smoke Box, inches . . . M	16 3/4	16 3/4	18	18	20	20
Width Breeching Connection, inches . . . N	10	10	10	10	12	12
Length Breeching Connection, inches . . . O	42	42	48	48	58	58
Height Steam Supply, inches . . . P	102 3/4	102 3/4	109 1/4	109 1/4	115 1/4	115 1/4
Height Return, inches . . . Q	22 3/4	22 3/4	23 1/4	23 1/4	23 1/4	23 1/4
Number of Common Brick	1100	1170	1230	1300	1350	1430
Outside Surface to be Covered, square feet	250	280	290	310	315	345

TABLE 13

SPECIFICATIONS—KEWANEE FIREBOX BOILERS—PORTABLE TYPE

See Figs. 15 and 16

Alternate Sizes Omitted, see Table 10

Number	0000	401	403	405	407	409	411	413	415	416	417	418	419	420
Diameter Boiler........inches	30	36	36	42	48	48	54	60	60	60	66	66	72	72
Length Boiler..feet and inches	5-5 1/8	5-11 1/8	7 1/8	7-8 1/8	8-3 1/4	10-9 1/4	10-9 1/4	12-1 1/4	13-11 7/8	15-9 7/8	15-6	16-11 7/8	15-7 7/8	17-1 7/8
Width of Firebox......inches	18	24	24	30	36	36	42	48	53	53	59	59	65	65
Length of Firebox......inches	20	26	38	38	38	50	50	56	56	62	62	68	68	74
Height of Firebox......inches	28	35	35	41	43	45	45	49	49	49	52	52	54	54
Number of Direct Tubes.....	10	17	17	21	30	30	40	29	29	29	38	38	48	48
Length of Direct Tubes.inches	30	30	31	39	42	60	60	70	92	108	102	114	96	108
Size of Direct Tubes....inches	3	3	3	3	3	3	3	4	4	4	4	4	4	4
Number of Return Tubes.....	12	18	18	25	30	30	36	30	30	30	33	33	48	48
Length Return Tubes...inches	53	59	72	80	83	113	113	129	152	174	168	186	168	186
Size of Return Tubes...inches	3	3	3	3	3	3	3	4	4	4	4	4	4	4
Heating Surface...square feet	75	122	151	219	281	375	477	583	701	804	905	1003	1003	1334
Area, Grate.......square feet	2.5	4.3	6.3	8	9.5	12.6	14.7	18.8	20.7	22	25.5	27.9	30.8	33.5
Diameter Breeching....inches	12	16	16	16	20	20	22	26	26	26	28	28	32	32
Diameter Stack.........inches	12	14	14	14	18	18	20	24	24	24	26	26	30	30
Minimum height of Stack feet	35	40	40	40	50	60	60	70	70	70	70	80	90	90
Diameter Stack 2 Boilers inches	14	18	18	20	24	24	26	32	32	32	34	34	40	40
Minimum height of Stack 2 Boilers...............feet	40	40	45	45	50	60	60	70	80	80	80	90	90	90
Size of Steam Opening (one) inches	2 1/2	3	3	4	6	6	6	7	7	7	8	8	8	8
Size of Return Opening (one) inches	2	2 1/2	2 1/2	3	4	4	4	5	5	5	6	6	6	6
Size of Safety Valve Opening inches	1 1/4	1 1/2	1 1/2	2	2 1/2	2 1/2	2 1/2	3	3 1/2	3 1/2	3 1/2	3 1/2	4	4
Number and Size of Supply and Return Openings for Water Boiler........inches	1-4	1-6	1-6	1-6	2-5	2-5	2-6	2-7	2-7	2-7	2-8	2-8	2-10	2-10
Height of Water Line...inches	53	66	66	69	72	72	78	83	86	86	90	90	97	97
Height Floor to Top of Shell inches	59	68	68	79	84	84	92	97	101	101	107	107	113	113

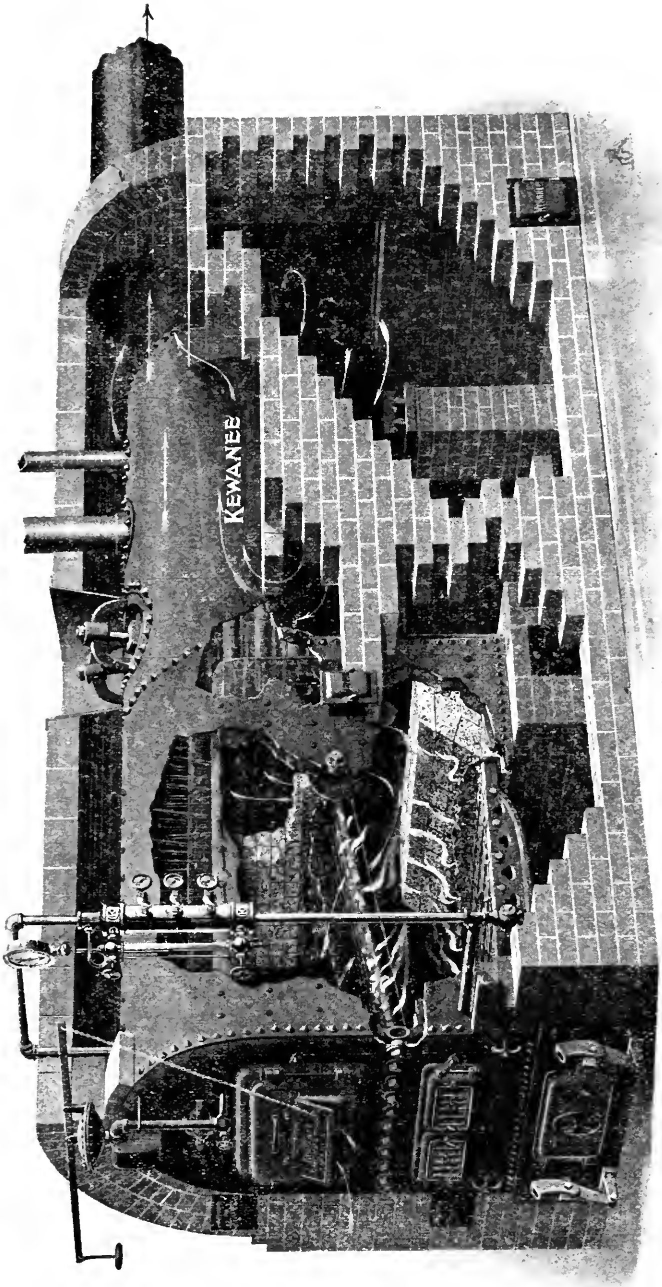


Fig. 17. Kewanee Smokeless Firebox Boiler—Brick Set

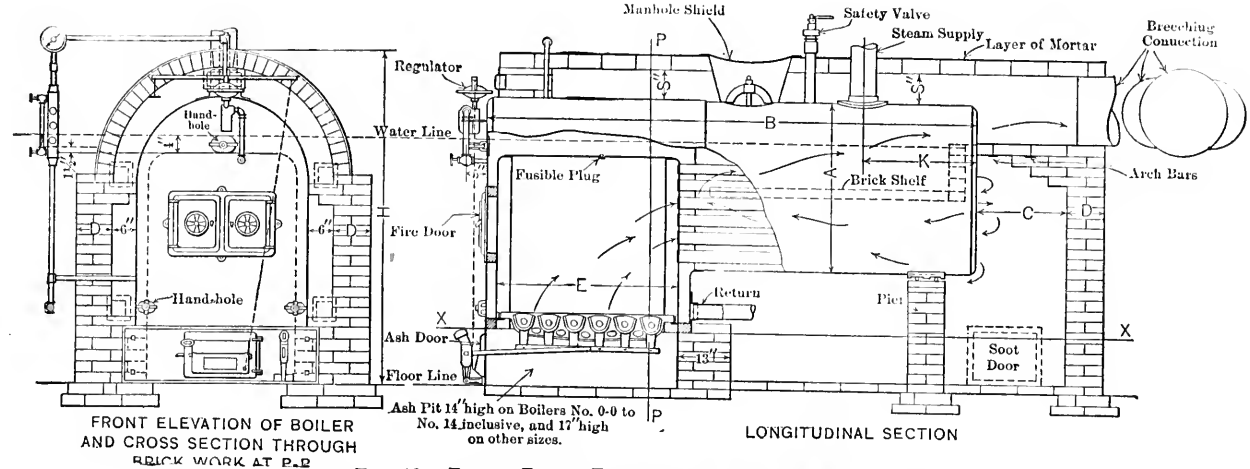


FIG. 18. DIRECT-DRAFT FIREBOX BOILER—BRICK SET.
For Dimensions and Capacities of these Boilers, see Manufacturer's Catalog.

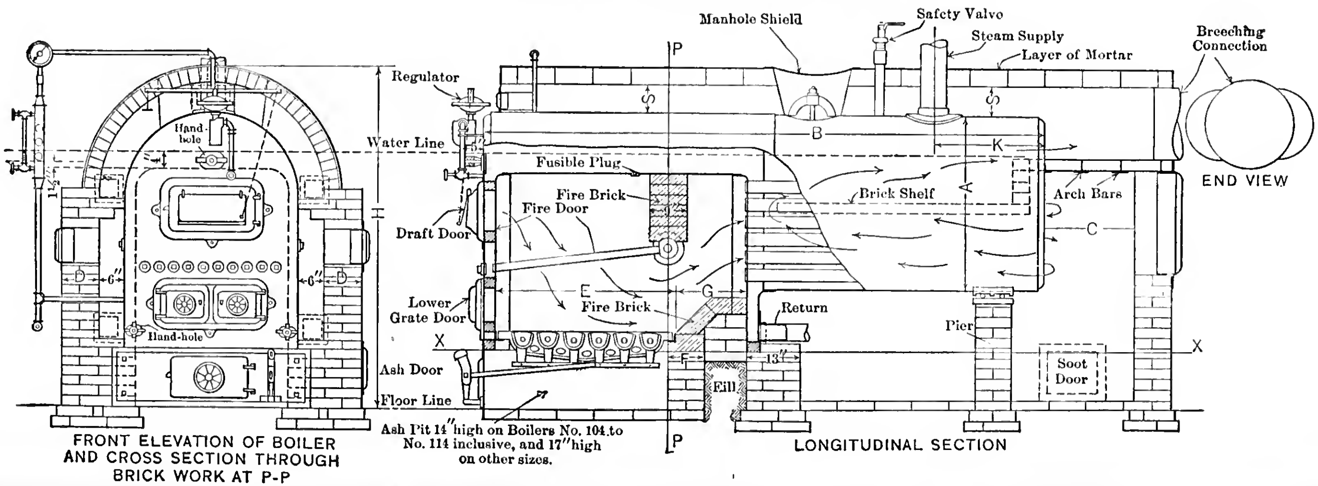


FIG. 19. SMOKELESS OR DOWN-DRAFT FIREBOX BOILER—BRICK SET.
(See Table 14.)

TABLE 14

SPECIFICATIONS AND DIMENSIONS KEWANEE SMOKELESS FIREBOX BOILERS

(See Fig. 19) *Kewanee Boiler Co.*

Number	104	106	108	110	112	114	116	118	120
Capacity, Steam, square feet	1600	2200	2900	3800	5000	7000	9500	12000	15000
Capacity, Water, square feet	2600	3600	4700	6200	8200	11400	15500	19600	24500
Diameter Boiler "A," inches	36	36	42	48	48	54	60	66	72
Length Boiler "B," feet and inches	8–7	11–7	11–4	12–4	15–4	18–4	20–4	20–4	20–4
Rear Space "C," inches	17	17	22	22	22	24	24	24	28
Thickness Wall "D," inches	9	9	9	9	9	13	13	13	13
Length Grate "E," inches	31	43	43	43	55	61	67	67	73
Width of Ash-pit "J," inches	31	31	37	43	43	49	54	60	66
Thickness Bridge Wall "F," inches	9	9	9	13	13	18	18	18	18
From Grates to Tube Sheet "G"	14	14	17	23	23	23	29	29	29
Total Height "H," inches	76	76	82	89	89	95	107	113	119
Location Supply "K," feet and inches	0–11	2–5	2–4	2–6	3–7	4–5	5–2	4–8	4–5
Top Flue Space "S," inches	7	7	7	8	8	8	10	10	10
Total Length "L," feet and inches	10–7	13–9	13–11	14–11	17–11	21–5	23–5	23–5	23–9
Total Width "W," feet and inches	5–6	5–6	6	6–6	6–6	7–8	8–2	8–8	9–2
Total Width "R," feet and inches	10–3	10–3	11–3	12–3	12–3	14–3	15–3	16–3	17–3
Common brick for one boiler	2500	2900	3400	4050	4550	6700	7900	8600	9600
Common brick for two boilers	4200	4950	5900	7000	7850	11800	13950	15200	17050
Fire brick for one boiler	72	72	90	108	108	150	190	230	300
Fire brick for two boilers	144	144	180	216	216	300	380	460	600

NOTE.—Intermediate sizes have been omitted. See manufacturer's catalog for similar dimensions of capacities given below (Table 15).

TABLE 15

CAPACITIES OF SMOKELESS BOILERS OMITTED FROM TABLE 14

Number	105	107	109	111	113	115	117	119
Capacity, Steam	1900	2500	3300	4400	5800	8200	10500	13000
Capacity, Hot Water	3100	4100	5300	7200	9500	13400	17000	21000

Horizontal Return Tubular Boilers. The horizontal return tubular boilers for heating service are usually made of the same material, mild steel, as is used in the firebox boilers. These boilers are very generally installed in the larger plants, and can be readily obtained in almost any part of the country. They are simply steel cylinders, provided with heads or "tube sheets," into which the fire tubes are "expanded."

The *furnace is exterior* to the boiler, and is usually placed directly below the front third of the shell, which is enclosed in a brick "setting," so that the hot gases of combustion first pass to the rear along the bottom half of the shell, and are then turned back by the setting, and return through the fire tubes to the front of the boiler or "uptake" from which they pass to the stack.

These boilers are ordinarily built for 100 lb. working pressure, although if so specified they will be designed for 150 or 200 lb. working pressure.

Tables of *dimensions* and rated *capacities* will be found in the chapter on "Power Boilers" in Section 2. *Capacities* of horizontal return tubular boilers are usually *stated in boiler horsepower,* and the rating is arrived at by measuring the heating surface (heating surface = surface with fire and hot gases on one side, and water on the opposite side), and allowing 10 or 12 sq. ft. of this surface for each boiler horsepower. The latter figure is safe and should generally be used in computing the ratings of these boilers.

The *selection of a brick set horizontal return tubular boiler* is usually not considered for heating service until the boiler capacity required exceeds 3,000 sq. ft. of radiation, or 25 to 30 horsepower.

The following method of proportioning these boilers for heating work is practised by the *U. S. Treasury Depart.* with satisfactory results.

R = total direct radiation in building in sq. ft., $b. h. s.$ = boiler heating surface in sq. ft., G = grate area in sq. ft. Then: $b. h. s. = R/7$ or 8 (*Carpenter* gives 7.2 or 9.6) for steam; or $R/11$ or 12 for hot water. Also $G = b. h. s./25$ for anthracite pea or rice coal; $b. h. s./30$ or 35 for bituminous coal plain grate, and $b. h. s./45$ for lower grate of down-draft furnace.

For *area of stack* see chapter on "Draft and Chimneys."

Tube area to be not less than $\frac{1}{8}G$ for anthracite pea and rice coal, and not less than $\frac{1}{6}$ lower grate for down-draft boilers and always larger than stack area.

The *maximum length of tubes* must not exceed 48 diameters, and tubes an odd number of feet long are not used. The maximum length of boilers 54″ diameter and less must not exceed 3 diameters, and for boilers over 54″ diameter must not exceed $2\frac{1}{2}$ diameters.

The following *check rule* based on a climate similar to that of New York city may be used to ascertain boiler horsepower for direct heating: In the average building allow 100 boiler horsepower for each 1,000,000 cu. ft. of contents for zero weather, and in moderately severe winter weather about $\frac{2}{3}$ of this will be required. Records at New York city indicate that from 40 per cent to 50 per cent of the maximum boiler horsepower demand will be required for the full heating season of 5,700 hours, or 8 months. In Federal buildings with a heating and ventilating apparatus the average ratio is 7000 cu. ft. of contents for each boiler horsepower.

For data on coal consumption see chapter on "Fuels and Combustion."

Cast-Iron and Steel Heating Boilers Compared. A comparison of cast-iron and steel or wrought-iron heating boilers should include the following features:

(1) The *relative efficiencies* as shown by tests indicate that the steel boiler, especially in the larger sizes is probably the more efficient; this is especially true of the down-draft type. The heat transmitting efficiency of the two materials, in the thicknesses used in boiler practice, is about the same so that the steel boiler owes its thermal superiority to design and possibly the effect of a heavier brick setting rather than to the nature of the material.

(2) The *serviceability* of the steel boiler is of marked advantage wherever the character of the attendance may result in abuse or neglect, since the cast-iron sections may crack, due to low water, freezing, etc. Injury would also be done to the steel boiler under these conditions, but it would not be serious enough usually to cause a shut down. It should be noted, however, that with *proper attendance and usage* the cast-iron boiler will undoubtedly outlast any steel boiler made.

(3) The *ease of installation* generally favors the cast-iron boiler, since its sections are so designed as to be readily carried through any ordinary door or window that is not less than 2′ 6″ wide. It should be remembered that this advantage is somewhat offset by the fact that great skill is required of the steam fitter in *assembling* these sections, whereas the steel boiler arrives as a complete unit and has only to be placed in position.

(4) The *extension of the system* may be necessary and more boiler capacity demanded. In this case the advantage is all with the cast-iron boiler provided sufficient additional sections can be added to give the required capacity. Practically nothing can be done with the steel boiler, except to replace it or add a second boiler.

(5) The *relative cost* of steel boilers is greater than that of cast-iron boilers of the same capacity, and in the smaller sizes the difference is very marked. Steel boilers are all assembled when bought, whereas the cast-iron boilers must be erected by skilled labor, and when this cost is added to purchase price the cast-iron boiler in the larger sizes is but little cheaper than the firebox boiler, even including the brick setting of the latter.

Selection of a Heating Boiler. The selection of a boiler for heating service must be governed very largely by the conditions under which the boiler must operate. These conditions include: kind of fuel, draft available, character of attendance, possibility of future extension, possibility of breakdown (reserve units), and, of course, the amount and character of the heating

load. In this connection the comparison of cast-iron and steel boilers just given should be referred to in making a selection.

The *kind of fuel* must be carefully considered, as, for example, a soft coal depositing large amounts of soot will render the indirect heating surface of a boiler almost useless. Hence boilers having a large proportion of direct surface and little indirect should be used, also all surface

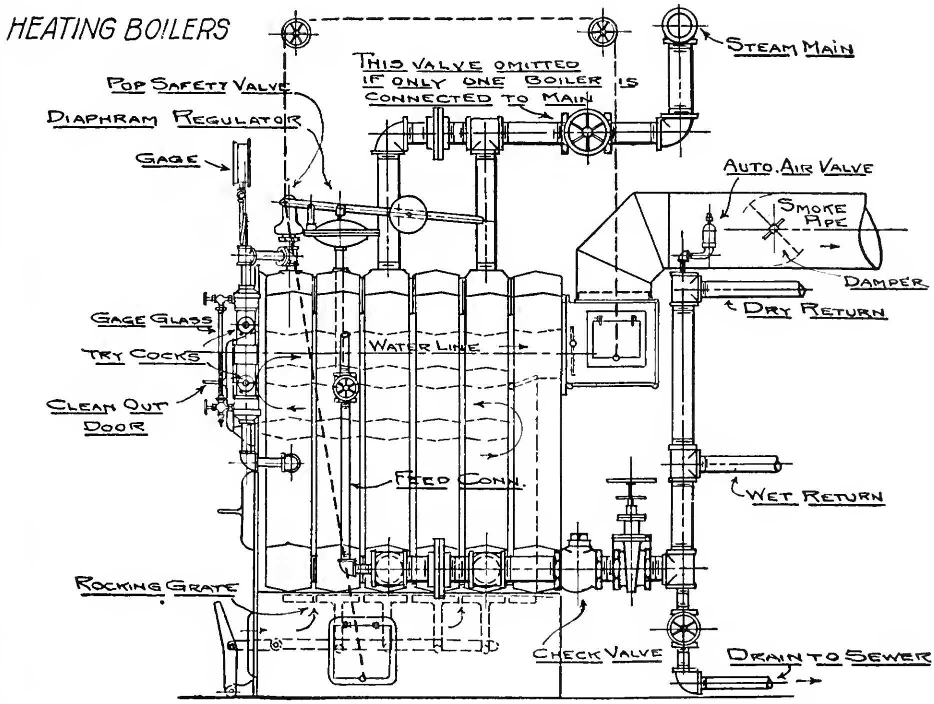


FIG. 20. TRIMMINGS AND CONNECTIONS FOR SECTIONAL STEAM HEATING BOILER

Connect each steam outlet of boiler to a common header as shown. Header to be same size as outlets. The returns may be connected to a common return header as shown or when there is only one return inlet on each side run branch to other side between check valve and boiler.

must be readily accessible for cleaning. With a smoky fuel down-draft boilers may be required.

The *draft available*, if limited, may absolutely preclude the use of certain boilers with tortuous flues in which the friction loss is high.

The *character of attendance*, especially in the case of public buildings, may require that a boiler capable of withstanding more or less abuse shall be installed. On the other hand, if a careful fireman is to operate the boiler abuse need not be taken into consideration.

The *possibility of future extensions* may require that a sectional boiler be used in the original installation, or that the piping and setting plans provide for future boilers. Unless such provision be made at the start it may prove unnecessarily expensive to add to the boiler equipment at a later date.

The *possibility of breakdowns* must always be kept in mind, and in hospital work and in similar institutions *reserve units* must be provided. This is often done by installing two units, each of ⅔ capacity so that by running one boiler, a unit more nearly proportioned to the average load will be available, and yet in case of failure of either boiler the building may be kept

under heat. In the larger plants, where power is generated as well as heat, at least one reserve unit must be provided to allow for cutting boilers out of service for cleaning.

Finally, the boiler must have the necessary *capacity to supply the amount and character of the radiation* or its equivalent, which is connected to it. This radiation may require a boiler for high or low pressure steam, or one designed as a water heater only. Cast-iron boilers should not be used for steam pressure in excess of 15 lb. or water pressures in excess of 35 or 40 lb.

TRIMMINGS AND CONNECTIONS FOR HEATING BOILERS

The trimmings and connections for heating boilers include certain standard fixtures and methods of fitting which in many states are prescribed by law.

Trimmings. The most common trimmings or fittings which are required on a low pressure *steam* heating boiler are: (1) *Safety valve,* (2) *Pressure gage,* (3) *Water column and gage glass,* and (4) *Damper regulator.* The fittings required for a *hot water* heater are: (1) *Altitude gage,* (2) *Damper regulator,* and (3) *Thermometers.*

The usual trimmings and connections for a sectional cast-iron steam heating boiler are shown in Fig. 20.

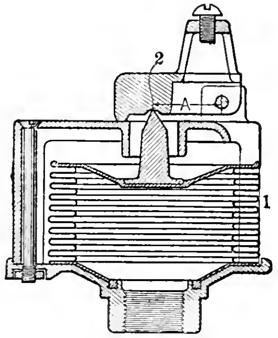


FIG. 21. SYLPHON STEAM OR VAPOR REGULATOR FOR DAMPER CONTROL.

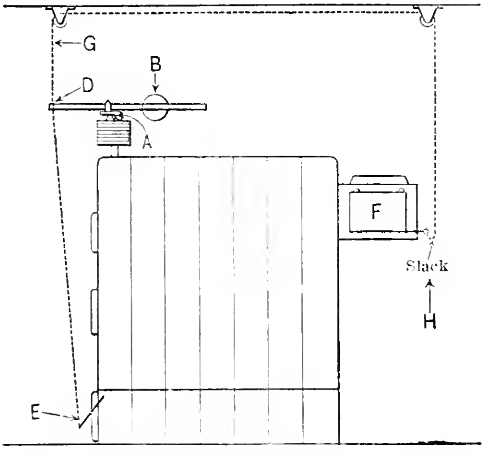


FIG. 22. DAMPER CONNECTIONS FOR SYLPHON STEAM REGULATOR.

Damper Regulators. Damper regulators (Figs. 21 to 24) are in almost universal use on both steam and water boilers, and they provide practically the only means of automatically controlling the rate of combustion in the boiler so that the heat generated in the furnace will be proportional to the heating requirements of the building.

These regulators are either for *steam of the diaphragm or bellows type* using the pressure existing in the steam space, or for *water of the thermostatic type* in which the temperature of the outgoing water acts upon the thermostatic element and causes it to open or close the drafts.

Sylphon Damper Regulator. This regulator is made for steam and water, is of the *bellows type,* composed entirely of metal, and is formed from two brass discs with accordion sides, made of steam brass of the best quality (Fig. 21). The accordion sides are formed of ten deep folds which permit of ample yet very sensitive expansion effect upon the vertical rod that connects the top of the bellows to the bar upon which the counterbalance weight is placed. The sides are not built up of separate discs, but are formed from a single piece of brass, so that there are no joints or seams to come loose and cause leakage.

The *Sylphon vapor regulator* is made similarly to the *Sylphon steam regulator* but with a bellows larger in diameter and of thinner metal. The distance A, between rocker pivots, is longer than in the latter. These features give extreme sensitiveness without impairing its durability. Point 2 is a knife-edge bearing (Fig. 21).

This regulator is specially designed for steam and vapor heating where very low pressures are used. It is extremely sensitive, and by means of double balancing weights can be adjusted

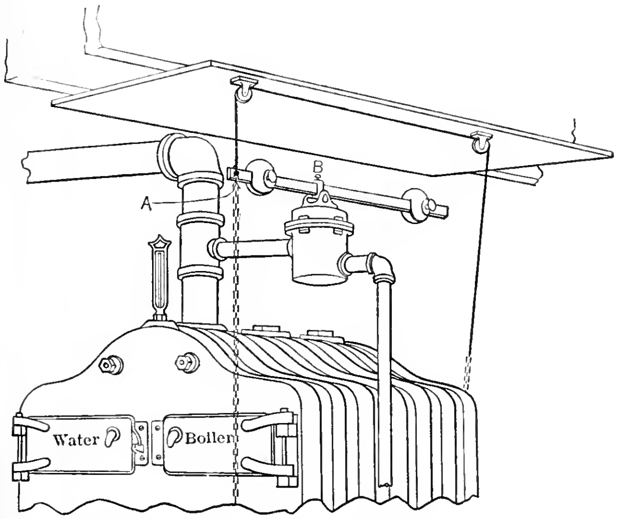


FIG. 23. WATER REGULATOR—SEPARATE INLET AND OUTLET.

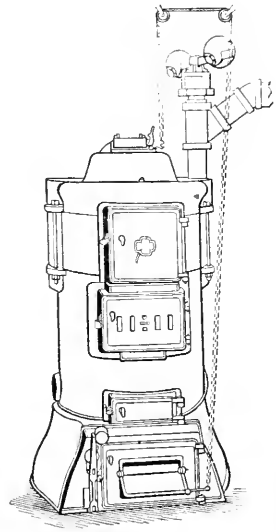


FIG. 24. WATER REGULATOR—SINGLE INLET.

to operate, if desired, exactly at atmosphere or any pressure up to 3 pounds. Tests show that it will operate under pressures measured in ounces.

The *method of installation and adjustment* (Fig. 22) of the *Sylphon* regulator is as follows:

After the boiler is set up and under fire raise whatever steam pressure it is desired to maintain, say 2 pounds.

When the gage shows 2 pounds, adjust the weight, B, on the regulator lever so that the chain connecting tilting draft damper, E, and check draft damper, F, is just taut, both dampers being closed. Then when the front draft, E, is open a little and check draft, F, closed, there will be a little slack in the chain as shown at the right or vice versa. The slack naturally comes in stretch of chain along ceiling, but slack at H shows relative amount.

If any greater pressure is generated, the check damper, F, will open and check the combustion; the pressure will lower to 2 pounds; then the check damper will close. If the fire is clean, the pressure will gradually increase without opening the tilting draft, E, and the regulator will operate the check draft only until the grate becomes covered with ashes; then the pressure will drop below 2 pounds, and the regulator will operate the tilting draft door on the ashpit.

The chains should never be disconnected from the doors, and any adjustment of pressures should be regulated by moving the weight, B, on the lever. From fulcrum, A, to end of lever, D, should never be less than 18 inches.

The *Sylphon regulator for water* is of the thermostatic type, and is made in two styles as shown in Figs. 23 and 24, and the following description of the regulator having separate inlet

and outlet will apply equally to the single inlet type, which latter is made up for only one temperature range.

This instrument is placed on the boiler or heater and automatically adjusts the dampers according to changes in water temperature. The water circulates through the regulator around an inner shell, similar to that used in the steam regulator, containing a volatile fluid. As the temperature increases the fluid expands a *Sylphon* bellows, tilting the lever and moving the dampers. This operation is reversed as the water cools. Weights are set to maintain different temperatures as desired.

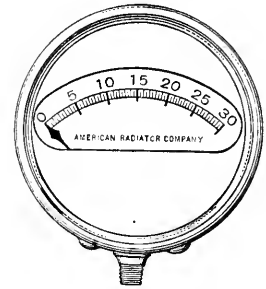


Fig. 25. Steam Gage.

Gages. *Pressure gages* (Fig. 25) are required on all *steam boilers*, and in case the boiler is to supply a vapor or vacuum system a *compound pressure and vacuum gage* is necessary.

These gages should be attached directly to the boiler or the top of the water column with a suitable *siphon or water seal* between the steam space and the gage so that hot steam cannot enter the mechanism of the gage and due to its heat expand the parts, thereby causing an incorrect reading. A shut-off cock should always be placed in this connection to allow for removal of gage for testing and inspection.

The pressure-indicating system consists of a curved hollow tube, known as a *Bourdon* tube, of oval cross section into which the steam forces first air and finally water from the siphon. As a consequence, the curved tube tends to straighten out, and in doing so causes the segment of a gear to engage and revolve a small pinion mounted on the same shaft as the indicating hand or pointer which travels over the face of the gage an amount proportional to the pressure exerted.

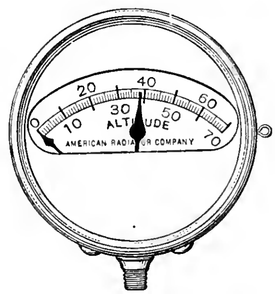


Fig. 26. Altitude Gage.

Gages for steam-heating boilers register from 0 to 30 pounds pressure per sq. in., as shown in Fig. 25, and may be equipped with a special crank movement and slotted face as shown, or may have a revolving hand and open face, like a clock. These gages usually have a cast-iron or pressed steel case fitted with a glass dial, held in place by a brass ring or armored steel frame, about 4½ inches in diameter.

Compound or pressure and vacuum gages register both above and below atmospheric pressure with a range from 15 pounds pressure to 30 inches vacuum; the negative pressures being usually expressed in inches of mercury. Their construction and operation are similar to the ordinary steam gage.

Altitude gages (Fig. 26) are required on all hot-water heaters and are also similar to the steam gage in principle and operation, except that no siphon is required and the pressure scale reads in feet of water column instead of pounds per sq. in. An adjustable red hand is usually provided for use as follows: When water is at its proper level in the expansion tank, remove front cover and glass, and set adjustable hand at pressure indicated by working hand; whenever pressure falls below this point, water should be added. This makes it possible to keep the system properly filled by simply observing the altitude gage at the " boiler " or heater.

Water Column and Gage Glass. A water column and gage glass with at least *three try cocks* on the former, as shown in Fig. 27, should be attached to the front of every steam heating boiler. This column should be so placed that the normal boiler water line will coincide with the middle of the gage glass and the middle try cock. Small heating boilers often have but two try cocks on the water column. The water column is usually of cast iron and is tapped to receive the above mentioned cocks and connections for the gage glass. In addition to this the column has a connection at the top to the steam space and at the bottom to the water

space of the boiler so that the water level in same is identical with the water level in the boiler.

This water level is visible in the *gage glass*, but in case of accident to the latter may be ascertained by use of the gage or try cocks on the water column itself. The valves at the top and

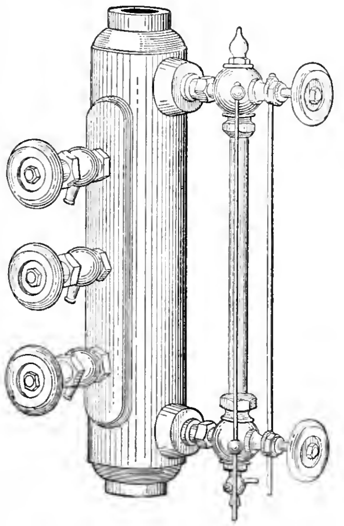


FIG. 27. WATER COLUMN GAGE GLASS AND TRY COCKS.

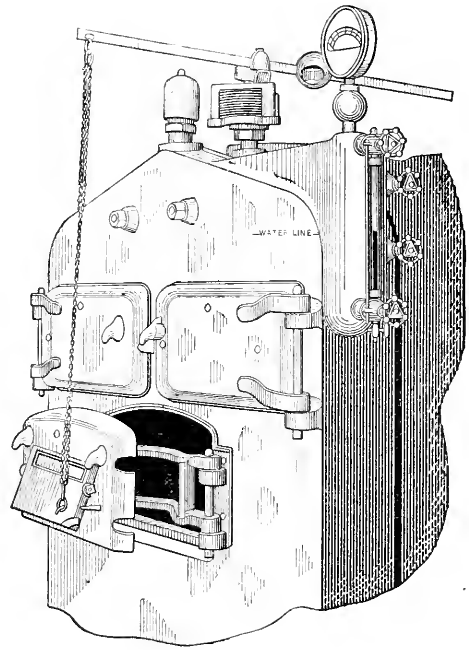


FIG. 28.

bottom of the gage glass may be closed in order to cut it out of service, and allow of replacement, and are often made *self-closing for power boilers* or those working under pressure. The bottom housing of the gage glass is also provided with a *drip cock* so that the contents of the glass may be blown out occasionally by opening this cock. The glass should always be protected by suitable *guards* made of 2 or 3 brass rods. Water columns for cast iron boilers are sometimes made *integral with the front section* (Fig. 28), but they are fitted up in the same manner as the *separate column* already described.

Safety Valves. The safety valves in use on heating boilers at the present time are usually of the spring closing type (Fig. 29) provided with an external hand lever for manual operation and testing. These valves are especially designed for low pressure service, and can be set for pressures up to 20 pounds, although usually set for 15 pounds working pressure. The *Massachusetts District Police* require a lock up type of valve which can only be set by the inspector or some one holding the key. These valves are set at 15 pounds, and then locked and sealed.

Safety valves for heating boilers range in size from ¾″ to 3″ and are threaded with a standard pipe thread. The valve itself is usually of brass, with a cast-iron base, and may be furnished in plain brass finish or nickel plated.

The *size of safety valve* varies with the boiler capacity, and in general the manufacturer equips his boiler with a valve from one to two sizes too small. That is to say, the valve could not relieve the boiler of steam at the proper rate to prevent the pressure from rising above 15

pounds if there was no other outlet for steam. For proper sizes of safety valves see list of tappings for steam boilers (Table 16). In case a tapping of suitable size has not been provided on the boiler itself, the valve may be mounted on a tee just above the main steam outlet, with no valve between it and boiler.

The *use of safety valves* is only necessary on steam boilers, as all low pressure water heaters are vented through the expansion line to the expansion tank. In case of special hot water systems of the closed type it may be necessary to equip the " boiler " or heater with a safety valve.

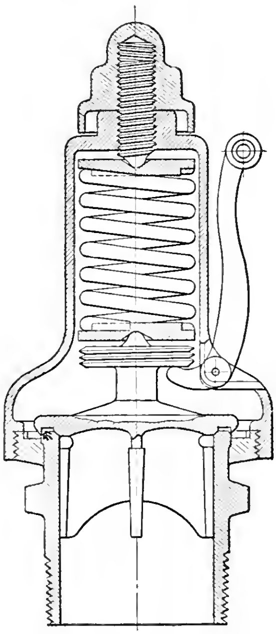
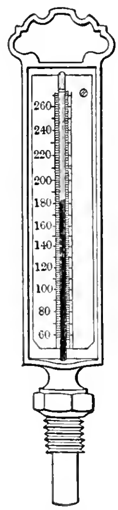
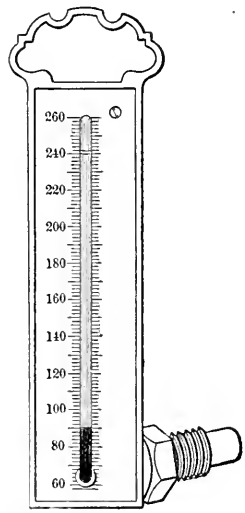


FIG. 29. SPRING ON POP SAFETY VALVE.



Straight.



Angle.

FIG. 30. FIG. 30*a*.
HOT-WATER THERMOMETERS WITH SEPARATE MERCURY BATH.

The fireman or attendant should *test* the safety valve at least *once a day* by lifting it with the hand lever provided for this purpose.

Other types of safety valves, making use of (1) *a dead weight* or (2) *a lever with weight* for closing the valve, have been used in the past, but are seldom specified in present day practice. These valves are easily tampered with and often with more or less serious consequences.

Thermometers. Hot water thermometers are required on hot water " boilers " or heaters, and are either attached directly to the boiler near the flow outlet or else are tapped into the flow-line itself close to the boiler. It is good practice to place one thermometer on the main flow and the other on the main return line. These thermometers usually read from 60° to 260° F., and are preferably of the *separable stem type* with mercury in the thermometer well, as shown in Figs. 30–30*a*. The well, which is threaded at the upper end to fit the tapping in the boiler or pipe, must be fully immersed in the hot water, and subject to the continuous circulation of same. The glass bulb of the thermometer is immersed in the mercury in the well and indicates its temperature.

Two patterns are ordinarily made, known as *the straight* and *the angle type* as shown, the latter being for use on risers, and both are of the separable stem type allowing for easy installa-

tion and adjustment. If the thermometer does not face in the right direction when screwed up tight, slightly loosen the small screw which is found on one of the faces of the hexagon nut and (without lifting) turn top of the frame to the desired position, after which tighten the screw.

Heating Boiler Connections. The connections which must be made to both *steam* boilers and *hot-water* heaters are: (1) *Steam or flow connections* from the top, (2) *Return connections* to the bottom, (3) *Blow off or drain connections,* (4) *Cold water connections* for filling. The boiler connections shown in Figs. 31 to 40 provide for both steam and water heating boilers and indicate approved methods for making main steam and flow connections as well as main return connections, blow-off or drain connections, and cold water connections. *Starting pipes* for water and *drip pipes* for steam are also indicated.

The steam or flow connections and also the return connections must be carefully designed to provide for *expansion* of the piping, as well as of proper *size* and *number* so as to give low outlet velocities, and uniform distribution of the return water over the boiler heating surfaces.

The *velocity of flow through the outlets* of low pressure steam heating boilers should not exceed 15 feet per second if entrainment of moisture is to be prevented, as with higher velocity the steam leaving the boiler may carry water with it.

It must be remembered that these boilers do not have " dry pipes " or " separators " within the boiler as do power boilers. Tables 16 and 17, following, give the tappings specified by the *U. S. Treasury Dept.* for heating boilers, and it will be seen that the average outlet velocity for steam is about 12 feet per second.

TABLE 16

STEAM BOILER TAPPINGS

Boiler Capacity, Sq. Ft. Rad.	Cu. Ft. Steam		Boiler Tappings		Velocity in Ft. per Sec.	Safety Valve Diam.
	Hour	Sec.	Steam	Return		
1200	6900	1.92	2-4″	1-2½″	11.0	2″
1500	8600	2.40	2-4″	1-2½″	13.4	2″
1800	10500	2.90	2-5″	1-3″	10.5	2½″
2200	12600	3.50	2-5″	1-3″	12.5	2½″
2600	15000	4.20	2-5″	1-3″	15.0	2½″
3000	17200	4.80	2-6″	1-4″	12.0	2½″
3600	20800	5.80	2-6″	1-4″	14.5	3″
4500	25900	7.19	2-6″	1-4″	17.9	3″

NOTE.—Volume of 1 pound of steam taken as 23 cu. ft. at approximately 2 lb. gage.

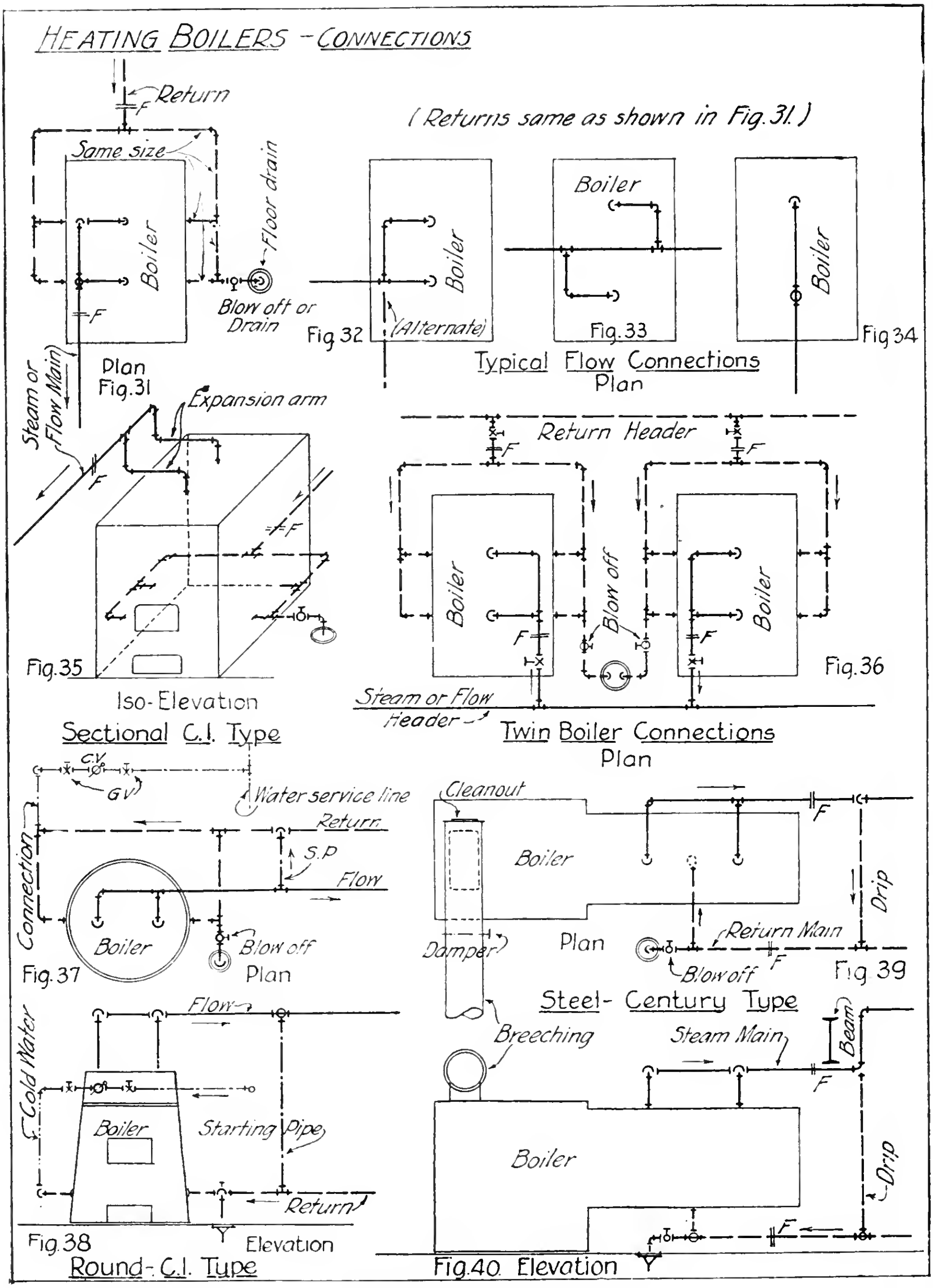
TABLE 17

WATER BOILER TAPPINGS

Boiler Capacity, Sq. Ft. Rad.	Boiler Tappings Flow and Return	Area Tappings, Sq. In.	Size of Main 150′ Run One Pipe	Area of Main, Sq. In.	*Safety Valve Diam.
2000	2-4″	24.4	1-5″	20.0	2″
2500	2-4″	24.4	1-6″	29.0	2″
3000	2-5″	40.0	1-6″	29.0	2½″
3600	2-5″	40.0	1-7″	38.0	2½″
1300	2-5″	40.0	1-7″	38.0	2½″
5000	2-6″	57.7	1-8″	50.0	2½″
6000	2-6″	57.7	1-8″	50.0	3″
7500	2-6″	57.7	2-6″	57.7	3″

* Use of safety valve optional usually on water boilers.

The *connections* shown in Figs. 31 to 36 are for a *cast-iron* sectional type of boiler, while those for a round cast-iron boiler are shown in Figs. 37 and 38, and for a *steel* boiler in Figs. 39 and 40. *Twin boiler connections* are made as shown in Fig. 36. Reference should also be made



Heating Boiler Connections.

Figs. 31 to 40.

to the figures in the chapter on "Direct Hot-Water Heating" showing expansion tank connections for water boilers.

Blow-off or drain connections must be provided for both steam and water boilers, and this connection should be made near the boiler and so arranged that the *entire system may be drained*

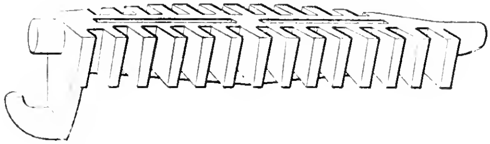


FIG. 41. REAR GRATE BAR, SHOWING STRONG TRUSSED CONSTRUCTION.

of water by opening the drain cock. (Figs. 31 to 40.) In the case of two or more boilers *separate blow-off connections* must be provided for each boiler on the boiler side of the stop valve on the main return connection. (Fig. 36.)

For heating boilers a *single plug cock* is usually satisfactory as a blow-off, although a gate valve may be installed next to the boiler in large installations in series with the cock, so that the wear due to opening and closing the blow-off may be taken by the valve and the cock will remain tight and prevent leakage. It is always advisable to have the blow-off discharge openly above a floor trap or sump so that leakage will be readily apparent, should the cock become worn. (Figs. 37 to 40.)

The blow off should be opened frequently for short periods while the plant is new, in order to draw off scale and sediment which naturally accumulates at the low point on the return side

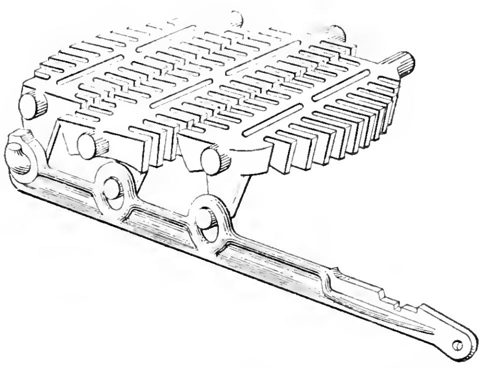


FIG. 42. GRATES ASSEMBLED, SHOWING CONNECTING BAR IN PLACE.

of the system. Subsequently the blow-off should be used occasionally only, except in spring and fall.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make up water to boilers. (Figs. 37 and 38.) This connection is usually of galvanized iron pipe, and is made to the return main near the boiler or boilers. The heating contractor is required to run this line to the nearest water service main, in which the plumber has left a plugged tee.

The line should be fitted with two standard gate valves, and one swing check valve, the latter to be in a horizontal run of pipe. The check valve is intended to prevent the pressure

in the boiler, if excessive, from backing up into the plumbing system of the building. Water connections range from $\frac{3}{4}''$ to $1\frac{1}{2}''$ in size, depending on size of boiler served by same.

Automatic water feeders are sometimes installed on the water connection in cases where all the steam is not returned as condensation, live steam being used for some purpose directly from the heating mains. These feeders operate under a float which rises or falls with the water level in the boiler.

Grates for Heating Boilers. Grates for heating boilers may be either *shaking* or *stationary,* and the former may be either of the *rocking* or *dumping* pattern or both. Shaking grates (Figs.

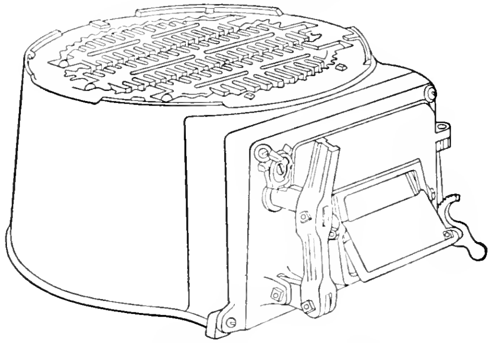


FIG. 43. NEW ARCO BASE COMPLETE.

41 to 46), which both rock and dump, are usually specified for portable heating boilers, while the stationary grates are often used in the larger brick set boilers of the return tubular type.

A grate is usually designed to allow the proper amount of air to pass for complete combustion, and at the same time prevent the fuel from dropping into the ash pit.

The *air space through grates* for heating boilers ranges from 50 to 55 per cent, and the maximum width of openings is varied from $\frac{3}{16}''$ for fine anthracite to $\frac{1}{2}''$ for the larger sizes. It is

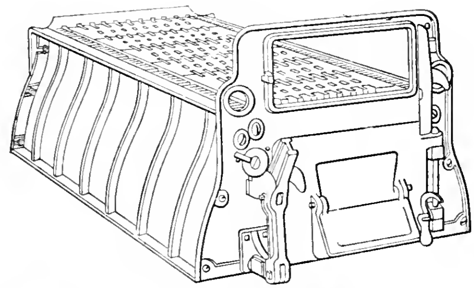


FIG. 44. VIEW SHOWING ASH-PIT AND THE ROCKING GRATES IN REGULAR POSITION.

always necessary to state in the specifications for the boiler whether hard or soft coal will be used on the grate, and to give the size of the coal as well.

Large shaking grates are divided into two halves so that either the *front* or *rear* half, or else the *right* or *left* half, can be shaken separately.

The Kewanee firebox boilers are equipped with a *herring-bone* type of grate (Figs. 45–46), having an air space of 55 per cent. This grate is also made in two sections for the larger sizes of boilers (Fig. 46) above 4500 sq. ft. of steam radiation, and the style shown in Fig. 45 is used

on the smaller boilers, although for boilers having a firebox longer than 38 inches the front half of this grate operates independently of the rear half.

Boiler Foundations and Ash Pits. Heating boilers of the portable type should be set on a substantial foundation of hard-burned brick set in cement mortar, and provision made for an

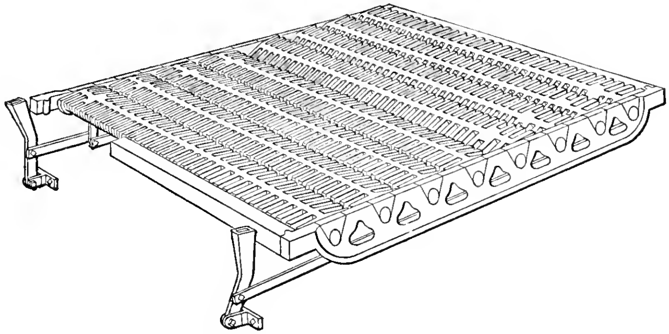


FIG. 45. HERRING-BONE TYPE OF GRATE.
For Smaller Sizes.

ash pit in the masonry as shown in Fig. 47. The side walls of the pit should be either 9″ or 13″ thick to serve as a bearing foundation, and the front of pit should be "ramped" as shown to facilitate the removal of ashes.

Such a pit as this will prevent any ordinary accumulation of ashes from obstructing the *free passage of air* through the grate bars at the rear end of the grate. This often happens when no pit is provided, with the resulting destruction of these bars which melt down, unless sufficient air is present to keep them cool.

Instructions for Operating Steam Heating Boilers. The operation of heating boilers re-

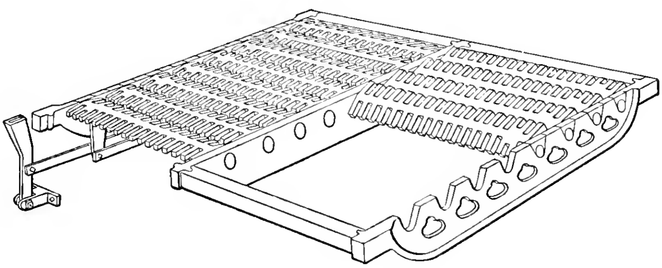


FIG. 46. TWO SECTION HERRING-BONE TYPE OF GRATE.

quires careful and regular attendance, and the following directions for firebox steel heating boilers by the *Kewanee Boiler Co.* apply almost equally well to any make of steam boiler:

Before starting the fire *fill the boiler with water until the water glass is about half full,* which should make the water line about 4 inches above the top of the upper row of tubes. Have the valves at the top and bottom of water glass open at all times; these to be closed only when it is necessary to replace a broken gage glass.

If there are any *valves* on the main steam supply pipe or on the main return be sure to have them *wide open* so the condensation can return to the boiler.

After the fire is well started *fill* the *firebox with fuel* to bottom of lower tubes, close the ash and fire doors. Carry as little *steam pressure* on the boiler as possible to heat the building properly. In cold weather steam will condense much faster than in mild weather, and it is neces-

sary to carry a greater pressure and make steam more rapidly. This can be regulated by adjusting the weights on the automatic *draft regulator* lever.

When *firing* close the draft door to avoid smoking. The most economical way to operate the boiler is to keep the entire grate surface covered with *a deep bed of fuel,* and allow the automatic damper regulator to control the fire. When a low fire is all that is required allow the ashes to accumulate on the grates to a depth of one to three or four inches, and when necessary to clean the fire of clinkers, do so with the poker and hook, but do not shake the grates except when necessary to increase the efficiency of the fire. When it is desired to make steam rapidly maintain a *clean fire.* To accomplish this, remove all clinkers and shake the ashes through the

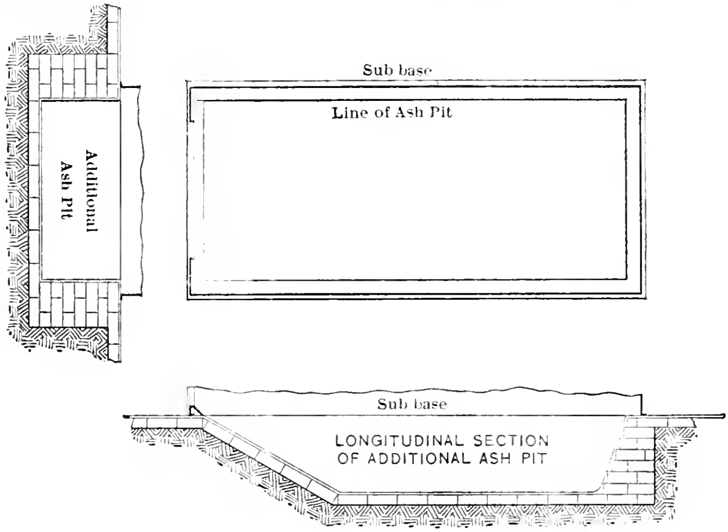


Fig. 47.

grates. Do not allow the ashes to accumulate in the ash pit, and if any hot coals are shaken through the grates into the ash pit it is well to remove them at once. There should be at all times at least *six inches of clear space below the grates.* This precaution will prevent burning and warping the grate bars.

The *chain* connecting the diaphragm lever to the *draft door* in the ashpit, and also the chain connected to the *check draft door* in the smoke pipe, should both be of such length as to allow both doors to be closed and chains taut when the lever is in a horizontal position. These doors should *never both be open at the same time.*

The *flues* should have all soot removed from them at least once every two weeks, or oftener if necessary, to keep them clean; this is to insure a free draft and the maximum possible heat transmission.

Draw off a small quantity of water every few days while the system is new, from the *blow-off cock* to remove any accumulated sediment. It is important to use clean water, as sediment in dirty water may in time cover the tubes and adhere to the shell to such an extent as to cause injury to the boiler, and reduce the heat transmission.

When the use of the apparatus is discontinued *in the spring,* water in the heater should be drawn off, but additional water turned into it until the water comes out of the safety valve, and it should be so left during the summer months. Before starting the fire *in the fall* drain the boiler by drawing all the water off and filling it again to the proper height as indicated by the gage glass. The glass should be kept slightly less than half full of water when the boiler is in operation.

CHAPTER VII

HEATING WATER IN TANKS AND POOLS

GOVERNING CONDITIONS

The type and size of apparatus best suited to heating water for domestic or manufacturing purposes depend upon: (1) the *amount* of water to be heated, (2) the *rate* at which it must be heated, gallons or pounds per hour, (3) the *range in temperature* through which the water must be raised, (4) the *efficiency*, or heat transmitting capacity, of the water heating surface, and (5) the *heating medium*, such as hot gas or steam.

Amount of Hot Water Required. In *domestic service* the amount of hot water required is usually based on the number of plumbing *fixtures* or *occupants* to be supplied, while for *commercial purposes* the amount depends on the particular process involved. An allowance in storage tank capacity of 20 gallons for each shower, 10 gallons for each sink and 5 gallons for each lavatory is customary in government buildings. In the case of shower baths from 2 to 3 gals. of hot water per minute per shower must be allowed or from 15 to 20 gals. per bath.

In *hospital service* an allowance of from 20 to 40 gals. of hot water per patient occupant per " day " is usually made. This water is used in from 10 to 12 hours, and hence for a 100-bed hospital it would be necessary to heat $40 \times 100 = 4000$ gals. per " day " or 400 gals. per hour from 50° to 180° F.

Temperature Range. Not only must the *quantity* of water be known but also the temperature *range* through which it is to be heated. The limits ordinarily set range from 50° F. at the inlet to 180° F. at the outlet, but for residences a somewhat lower outlet temperature of 140° F. to 160° F. may be used. It must be remembered that in domestic service both hot and cold faucets are usually opened at the same time, and that the resulting temperature must prove satisfactory. Hence a temperature above 150° F. is generally necessary.

In case a storage tank is used, a heater which will increase the water temperature from 25 to 30 degrees per hour is satisfactory for residences, but for apartment houses and small hotels, the increase must be from 40 to 45 degrees per hour, when using a storage tank. This means that the *hourly capacity* of the heater is much less than the capacity of the tank, so that the hot water demand on the latter *must not be constant*, but must permit of periods when the heater can " catch up " by working on the tank alone, as explained in the following paragraphs.

Demand Factor. It is apparent from the above that the rate at which the hot water is used determines the size and capacity of the apparatus for a given temperature range. Thus, if all the water is used in one hour a much larger heater is required than would be needed if the same amount was used in 3 or 4 hours, at the same temperature of supply. In consequence of this condition it is generally customary to provide a *storage tank* from which the hot water supply is drawn. This tank has a capacity much greater than the hourly capacity of the heater which can now be made small since it operates on the storage tank during periods when no hot water is being withdrawn, and thus maintains the desired temperature. In case the hot water demand is practically constant, a much larger heater suitable for continuous service must be installed although the *constant* rate of supply may be no greater than the *intermittent* rate provided for in the previous case.

Efficiency of Heating Surface. The heat transmitting capacity of any water heating surface varies (1) with the nature of the heating medium, (2) with the temperature difference between the heating medium and the average water temperature, (3) with the velocity of flow of the water over the surface, and (4) with the thickness and nature of the material through which

the heat is transmitted. Practical transmission values, applicable to water heating practice, are given under the discussion of each type of heater considered.

For any given condition as, for example, horizontal steam coils lying in a storage tank of water the heat transmitted per sq. ft. of surface per hour is:

$$H = K \times (t_2 - t_1) \times A,$$

where K = B.t.u. per sq. ft. per 1° difference in temperature per hr.
t_2 and t_1 = temp. of steam, and average temperature of water.
A = area of surface in sq. ft.

Values of K used by *Rietschel*, the German engineer, are as follows:
K for air or smoke to water through a cast-iron or steel plate = 2.6 to 4.0 B.t.u.
K for steam to water through a metal wall = 160 to 200 B.t.u.

TYPES OF HEATERS

Various types of hot water heaters are in use, ranging from coal or gas burning tank heaters, with which should be included water backs and firebox coils, to pipe coil heaters using either live or exhaust steam, or even steam at less than atmospheric pressure.

Tank Heaters. The use of *coal or gas fired tank heaters* is the most satisfactory way of heating water when there is no live or exhaust steam available. These heaters when coal fired are *rated* on a basis of grate area and rate of combustion in exactly the same manner as house heating boilers except that a somewhat better efficiency may be obtained due to the lower mean temperature of the water in the boiler.

If the quantity of water to be heated per hour and the temperature rise are known the *grate area* is readily found as follows:

$$G = \frac{W\ (t_2 - t_1)}{F \times E \times C}$$

G = grate area in sq. ft.
W = weight of water to be heated per hour in pounds.
t_2 and t_1 = leaving and entering water temperatures, usually 180° F. and 50° F.
F = heat value of the fuel in B.t.u. per 1 lb.
E = efficiency of heater, say 65 per cent.
C = rate of combustion per sq. ft. of grate per hour = 3 lb. for small grates, less than 2 sq. ft., and 5 lb. for large grates. This rate may become 8 lb. with frequent firing and constant attendance.

Example. What size of heater is required to heat 500 gals. of water from 50° F. to 180° F. per hour, using anthracite coal, having a heating value of 12,000 B.t.u. per pound.

$$G = \frac{500 \times 8\tfrac{1}{3} \times (180 - 50)}{12{,}000 \times 0.65 \times 5}$$
$$= 14 \text{ sq. ft.}$$

If in the above example the demand is intermittent, and for short intervals only, then a smaller heater may be used, on the assumption that it may be allowed 2 or 3 hours for heating the water in the tank. It is generally advisable to allow extra tank capacity in these cases so that a storage volume several times the maximum hourly demand is available. The *ratio of storage capacity to grate area* is often as high as 250 gals. per sq. ft., yet 1 sq. ft. of grate will only heat,

$$W = \frac{G \times F \times E \times C}{(t_2 - t_1)} = \frac{1 \times 12{,}000 \times 0.65 \times 5}{(180 - 50)} = 300 \text{ lb. or } 36 \text{ gals.}$$

from 50° F. to 180° F. when burning coal at a 5 lb. rate.

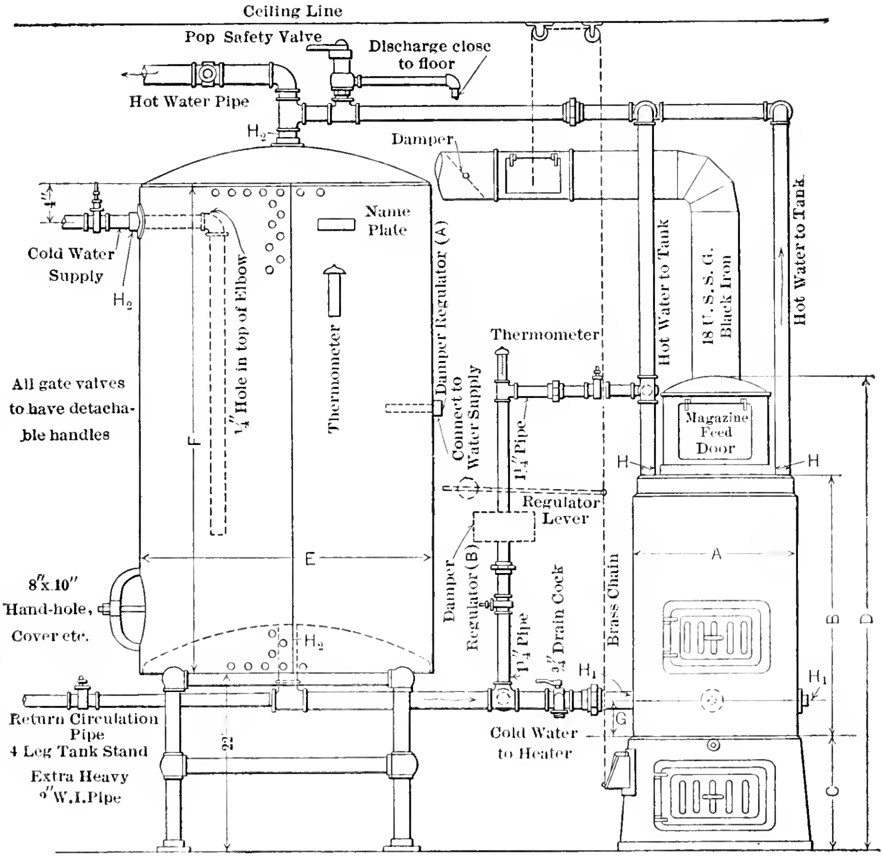


FIG. 1. STEEL TANK AND HEATER.
From General Specifications of the Treasury, War and Navy Departments.
See Table 1.

TABLE 1

DIMENSIONS OF TANK AND HEATER

Outfit Number		150A	200A	300A	400A	500A
Number of Heater		150S	200S	300S	400S	500S
Diameter of Heater	A	17"	21"	25"	25"	30"
Height of Heater	B	36"	30"	36"	48"	42"
Height of Base	C	14"	14"	14"	14"	16"
Height Over All	D	62"	57"	64"	76"	70"
Thickness of Outer Shell of Heater	....	3/16"	3/16"	3/16"	3/16"	¼"
Thickness of Inner Shell of Heater	....	¼"	¼"	¼"	¼"	¼"
Thickness of Head of Heater	....	⅜"	⅜"	⅜"	⅜"	⅜"
Bot. of Heater to Center of Opening	G	3"	4"	4"	4"	4¾"
Number and Size of Flow Openings	H	2–1½"	2–2"	2–2"	2–2"	2–3"
No. and Size of Return Openings	H_1	2–1½"	3–2"	4–2"	4–2"	4–3"
Number of Tank	....	36V	42V	48V	54V	60V
Size of Tank	E x F	36" x 60"	42" x 60"	48" x 60"	54" x 60"	60" x 60"
Thickness of Shell of Tank	....	¼"	¼"	¼"	5/16"	5/16"
Thickness of Convex Head of Tank	....	5/16"	5/16"	5/16"	⅜"	7/16"
Thickness of Concave Head of Tank	....	⅜"	⅜"	7/16"	½"	9/16"
Size of all Pipe Openings	H_2	1½"	2"	2"	2"	3"

Heater and Tank Equipment. A *steel tank heater* of the "magazine feed" type together with storage tank and all connections thereto is shown in Fig. 1, as specified by the *U. S. Government.* It will be noted that the heater is under the control of an automatic damper regulator which maintains a constant temperature of about 160° F. in the storage tank, by suitable control of the heater drafts.

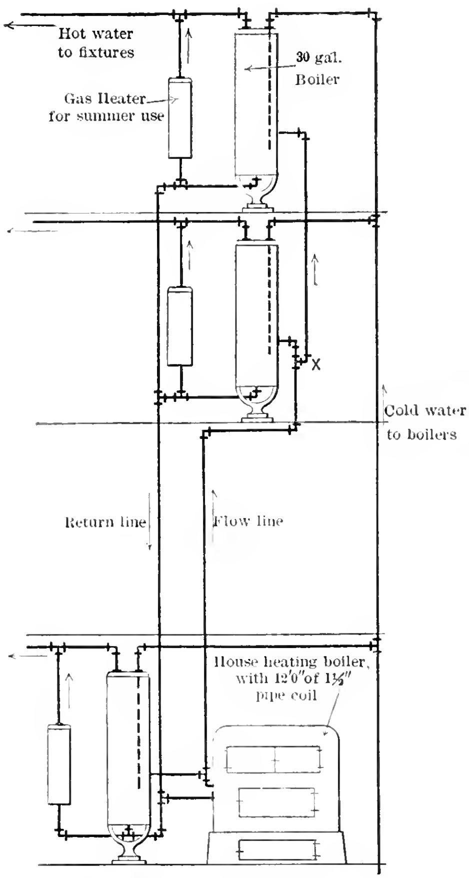


Fig. 2. Combined Gas Heater and Pipe Coil Installation.

These heaters have *fire-pots* 12″, 16″, 20″, 20″ (extra surface) and 25″ in diameter with a heating *capacity* of 150, 200, 300, 400 and 500 gals. of hot water per hour respectively. These heaters must *weigh*, including all castings, not less than 600, 700, 950, 1130 and 1500 lb. for each size, and must be *made of* flange steel of 55,000 to 60,000 lb. tensile strength. Each heater must be *tested* to 150 lb. hydrostatic pressure, and certificate of test furnished.

The *tank* is constructed of the same material and tested in the same manner as specified for the heater, with all openings suitably reinforced. A 1¼″ brass pop safety valve is provided, and set to open at 25 lb. above maximum water supply pressure.

Cast-iron heaters are also specified by the government for use wherever the water may have a corrosive action on the heater metal. These heaters must have *fire-pots* 12″, 15″ and 18″ in diameter, with a heating *capacity* of 150, 200 and 300 gals. of water per hour respectively, and must *weigh*, with all castings, not less than 400, 575 and 775 pounds each. The fire-pot must not be less than 14″ *deep* with *corrugated sides* to better withstand the water pressure. A hydrostatic *test* of 100 lb. per sq. in. is required both at the factory and after installation.

Water Backs and Pipe Coils. The use of *water backs* (or fronts) and pipe coils *placed directly in the firebox* or fire-pot of ranges, stoves, boilers, and furnaces is a very common means of providing hot water for domestic purposes. These water backs are usually cored castings forming one side of the fire-pot and are proportioned on the basis of 2 to 2½ sq. in. of heating surface for each gallon capacity in the hot water storage tank to which they *must be* always connected, using either ¾″ or 1″ flow and return piping.

The average *heat transmission of water backs* with water entering at 50° F. and leaving at 150° F. is approximately 12,000 B.t.u. per sq. ft. per hour, so that it will be readily apparent that 1 sq. ft. of this surface will only heat about $\frac{12{,}000}{8\frac{1}{3} \times (150 - 50)} = 14.5$ gals. per hour. Such surface, when used for house heating, is usually listed to supply about 75 sq. ft. of water radiation at a transmission value of 150 B.t.u. per sq. ft. per hour, or 11,250 B.t.u. With a bright fire,

however, the heat transmission may easily rise to 15,000 B.t.u. per sq. ft. per hour, although it is probably safer to use the lower value, and thus allow for effect of scale and sediment.

The use of *pipe coils* inserted into the firebox of steam and water boilers and furnaces is a common but unsatisfactory means of heating the domestic supply. These coils not only *reduce the capacity of the boiler* for house heating, but also *obstruct the fire-pot* and *chill the fire*, thus interfering with proper combustion. Furthermore, since the heat transmission of these coils increases with the fire intensity, the hot water supply will be *overheated* in cold weather and *underheated* in mild weather.

The boiler manufacturers specify that when a hot water coil is used, the radiation capacity of the boiler must be reduced by 1½ sq. ft. of steam radiation and 2½ sq. ft. of water radiation for each gallon heated. This allowance is too small, unless from 2 to 3 hours is allowed for heating each gallon used.

In contact with the fire these coils will give a heat transmission of 15,000 B.t.u. per sq. ft. per hour, but when placed above the fire the transmission may be much less depending on the intensity or brightness of the fire. It is therefore evident that each square foot of coil will heat about 18 gals. of water per hour from 50°F. to 150°F. since,

$$1 \times 15{,}000 = 18 \times 8\tfrac{1}{3} \times (150 - 50),$$

and common practice allows approximately 1 sq. ft. or 2 lineal feet of 1½″ pipe for heating 15 gals. of water per hour.

A typical installation of this sort is shown in Fig. 2. This installation provides for both summer and winter service since gas water heaters are connected to each boiler for use when the house-heating boiler is not in operation.

Gas Water Heaters. The use of *gas water heaters* is quite common in localities where either natural or artificial gas is available. These heaters are usually of the *instantaneous type*, so called because the coils or heating surfaces are made large enough to raise the water temperature from 50° F. to 100° F. on one passage through the heater, and discharge the heated water directly into the hot water service line (Fig. 2). These heaters have a high efficiency, usually in the neighborhood of 60–70 per cent, but may prove dangerous unless they are positively connected by means of a smoke pipe to a suitable chimney.

The capacity is usually stated in gals. per minute of water raised from 50° F. to 140° F. or 150° F., and the gas consumption is easily calculated as follows:

$$C \times H \times E = G \times 8\tfrac{1}{3} \times (t_2 - t_1),\quad G = \frac{C \times H \times E}{8\frac{1}{3}\,(t_2 - t_1)}$$

C = cu. ft. of gas burned per minute.
H = heat value of gas in B.t.u. per cu. ft., or 600 for artificial and 1000 for natural gas.
E = efficiency, approximately 70 per cent.
G = capacity of heater in gals. per minute.
t_2 and t_1 = final and initial temperatures.

The average gas water heater of the non-automatic type will burn from 35 to 40 cu. ft. of gas per hour, and will raise about 50 gals. of water from 65° to 100° F. in the same time with an efficiency of about 65 per cent when using gas of 650 B.t.u. per cu. ft. This service has been found too slow in practice and heaters capable of burning from 70 to 80 cu. ft. of gas are to be preferred.

The *large sizes of instantaneous heaters* such as the *Ruud* are usually made to operate automatically, and arranged for thermostatic control. In this way the gas consumption varies automatically with the rate of water flow through the heater, and the thermostat maintains a uniform and satisfactory outlet temperature of about 150° F. in the storage tank.

Tests of these automatic heaters show efficiencies as high as 70 per cent, requiring a consumption of from 1½ to 2 cu. ft. of ordinary illuminating gas per gallon of water heated, which with gas at 80c per 1000 cu. ft. makes it possible to heat 8 gallons of water 100° F. for 1 cent. The manufacturers will seldom guarantee more than 65 per cent heating efficiency for an automatic gas heater storage system.

The *Ruud Instantaneous Tank Heater* (Fig. 3) makes use of multiple heating coils of vari-

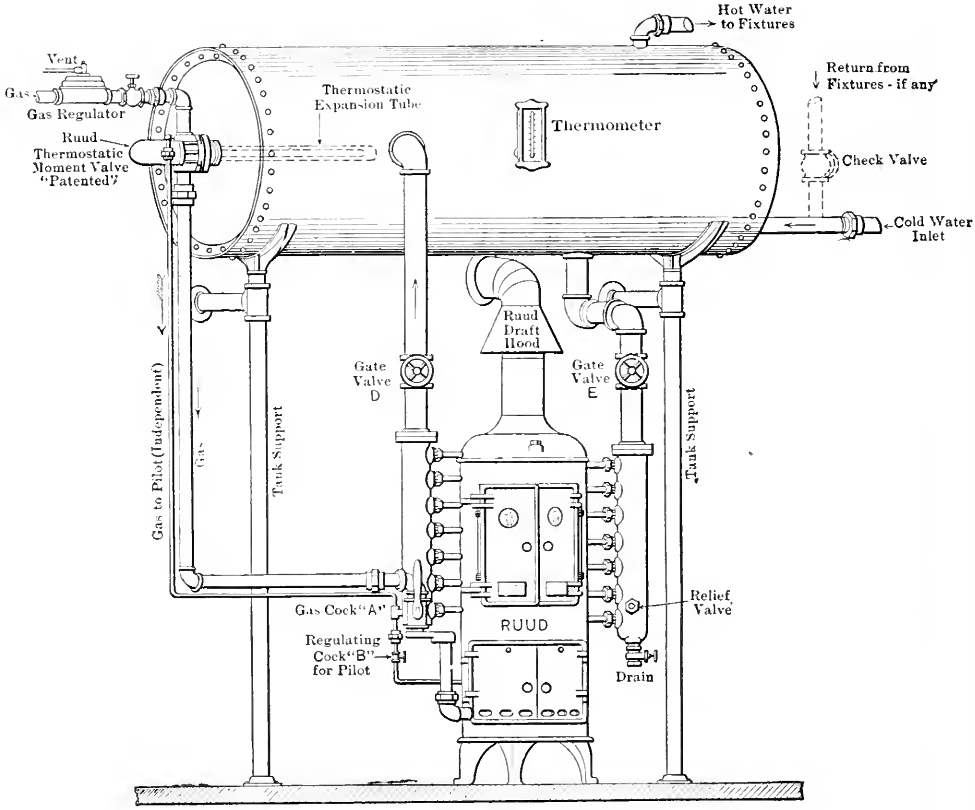


Fig. 3. Ruud Instantaneous Tank Water Heater.

able length and diameter as shown in Fig. 4. These coils form the heating elements just above the burners, with the coil of largest diameter and shortest length, therefore passing the most water, placed at the bottom where the greatest heat exists. Any coil is readily removable.

The thermostat, placed at the center of the tank, operates the gas valve with a "snap" action on a 25° range in temperature at this point, so that the gas is either *all on* or *all off*. Hence, once the entire tank has been heated to 150° F. and the gas all shut off, no gas is turned on until enough water has been used to cause the temperature at the center of the tank to drop 25° when the gas is turned on full and immediately ignited by the pilot light. In this way there is always a full flow of gas and no flickering at the burners, and the coil and burners always operate at full capacity for definite periods.

Sizes, dimensions, and water connections for these heaters are given in Table 2.

For maximum economy in operation all tanks must be covered with 2″ of 85 per cent magnesia covering.

TABLE 2

CAPACITIES AND DIMENSIONS RUUD TANKS AND HEATERS

Heating Capacity of Heater in Gallons per Hour	Size of Tank in Gallons	Tank Dimensions	Circulating Pipe	Hot Supply and Cold Inlet
100	100	5′ x 22″	1½″	1¼″
200	200	8′-6″ x 24″	2″	1½″
300	300	8′ x 30″	2″	2″
400	425	8′ x 36″	2½″	2″

NOTE.—Tanks from 50% smaller to 50% larger than the heater capacity per hour may be used if necessary.

Steam Coil Water Heaters. The hot water supply for large buildings is usually heated by means of submerged steam coils of copper, brass or iron placed in the hot water tank.

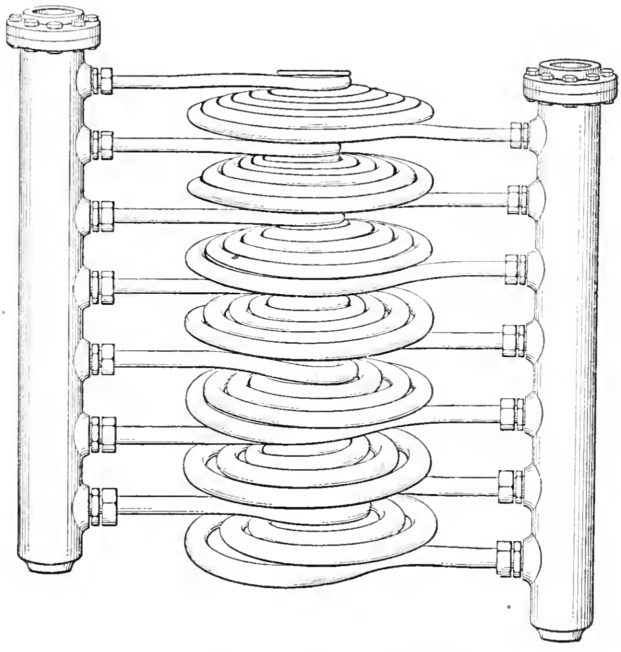


FIG. 4. HEATING COILS FOR RUUD HEATER.

Heat Transmission of Steam Coils. When steam coils are *surrounded by water* at a lower temperature the coefficient of heat transmission becomes vastly greater than when they are surrounded by air, due to the greater capacity of water for absorbing heat from the surface of these coils. This heat absorbing capacity may be still further augmented if the velocity with which the water flows over the surface of the tubes is increased by any means, such as the circulating pump in surface condenser practice.

Heat transmission values as high as 1,000 B.t.u. per sq. ft. per hour per 1° difference between the steam and average temperature of the water have been obtained with perfectly clean

tubes in condenser service. Since the tubes are not clean in actual practice, the above value is reduced to from 300 to 400 B.t.u.'s as the actual coefficient of transmission for use in designing a surface condenser, where circulation of the condensing water is maintained by a pump.

The heat transmission of *coils placed horizontally* in a water storage tank where *no mechanical circulation* is maintained is seldom greater than 500 B.t.u. per sq. ft. per hour per 1° difference between the steam and water with perfectly clean tubes. In this case the circulation is

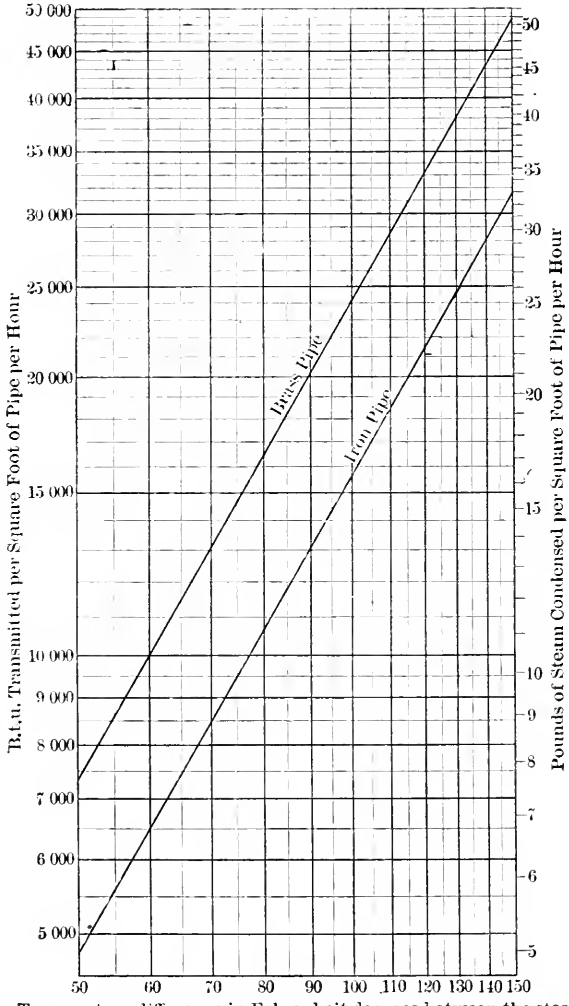


FIG. 5.

HEAT TRANSMISSION OF BRASS AND IRON STEAM COILS IN STORAGE TANKS.
(Tests on horizontal pipe in a tank of water.) *American Radiator Co.*

due almost entirely to natural convection currents, which of course are greatest for large temperature differences, so that the coefficient of transmission is a function of this difference, increasing with it.

Actual *tests made with brass and iron piping* lying horizontally in a tank of water show that the transmission efficiency of brass is superior to that of iron piping as indicated on the logarithmic chart shown in Fig. 5. This chart is based on the use of low pressure steam up to 10 lb. gage, and all transmission values have been reduced 50 per cent to provide for fouling of the coils and a consequent reduction in transmission efficiency in actual service. For example, the coefficient at 100° difference in temperature for brass pipe is 24,000/100 = 240 B.t.u. per sq. ft. of coil as read from chart, but with perfectly clean pipe the coefficient for this temperature range would be 480 B.t.u. All transmission values are based on external surface of coil. At 60° difference between the steam and water, the allowable coefficient is 170 B.t.u. based on a maximum of about 350 B.t.u. as determined by test with perfectly clean pipe.

The scales give both B.t.u. transmitted and pounds of steam condensed per sq. ft. of coil per hour for given temperature differences. Copper coils are somewhat more efficient than brass.

Amount of Coil Surface Required. The following examples show how to use the chart:

Example. It is required to condense 500 pounds of steam per hour in a pipe-coil immersed in the water of a storage tank. How many square feet of pipe should the coil contain?

Temperature of steam in pipe	220	degrees F.
Initial temperature of water	40	"
Terminal temperature of water	160	"
Mean temperature of water	100	"
Temperature difference steam and water	120	"

Observe that the line for *iron* pipe intersects the vertical line of 120 degrees temperature difference at the horizontal line representing 22.4 pounds. The intersection of the line for *brass* pipe shows 34.5 pounds.

The quantity of pipe required in square feet is determined by dividing the 500 pounds of steam which must be condensed per hour by the quantity of steam one square foot of pipe will condense.

For *iron* pipe, 500/22.4 = 22.2 sq. ft. would be required.

For *brass* pipe, 500/34.5 = 14.5 sq. ft. would be required.

Example. Suppose a tank containing 300 U. S. gallons of cold water at 60° F. is to be heated by low pressure steam, at a pressure of 5 lb., to a temperature of 140° F., in 2 hours; how many sq. ft. of *brass* pipe will be required, and how much steam will be condensed per hour?

300 U. S. gallons weigh 300 × 8.33	2,500	pounds
Total temperature rise desired	80	degrees F.
Temperature rise per hour	40	"
Heat required per hour = 2,500 × 40	100,000	B.t.u.
Temperature of steam at 5 lb. pressure (approximate)	227	degrees
Mean temperature of water	100	"
Mean temperature difference between steam and water	127	"

Where the line for *brass* pipe intersects the vertical for 127 degrees, read the transmission per sq. ft., 36,500 B.t.u., and the condensing power, 38 pounds of steam per sq. ft. per hour. The total sq. ft. of *brass* pipe required will then be 100,000/36,500 = 2.74 sq. ft. The condensation per hour would be 2.74 × 38 = 104.1 pounds.

Pipe coil *heaters are also proportioned* on the basis of 1 linear foot of 1″ iron pipe for each 5 gals. in the tank, on the assumption that the contents are to be heated from 50° F. to 180° F. per hour. This would mean that a 150 gal. tank would have 30 lineal ft. or 10 sq. ft. of pipe coil, and its heat transmission per sq. ft. would be, $H = 150 \times 8\frac{1}{3} \times (180 - 50)/10 = 16{,}250$ B.t.u

By reference to the chart (Fig. 5) it will be seen that with steam at 220° F. and average water temperature at 115° F., as given for this case, the heat transmission is 17,000 B.t.u. per hr.

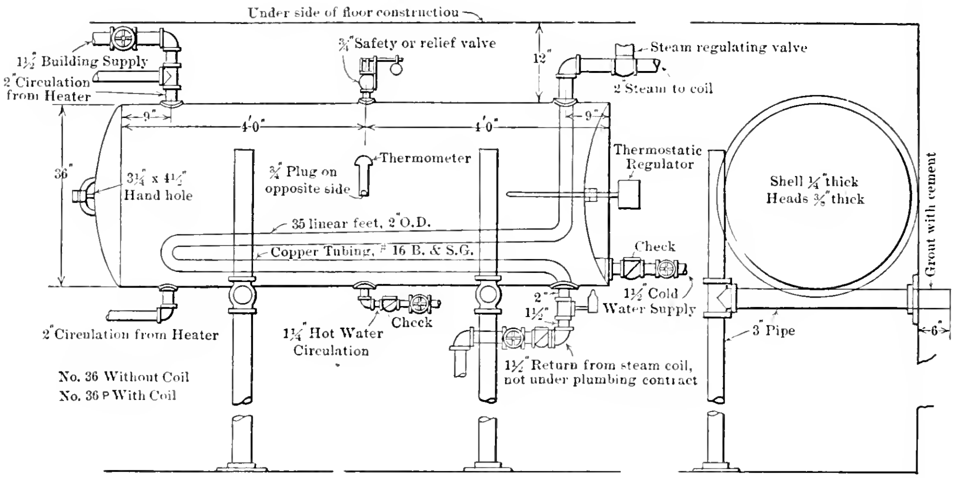


FIG. 6. PIPE COIL HEATER.
From General Specifications of the Treasury, War and Navy Departments.
See Table 3.

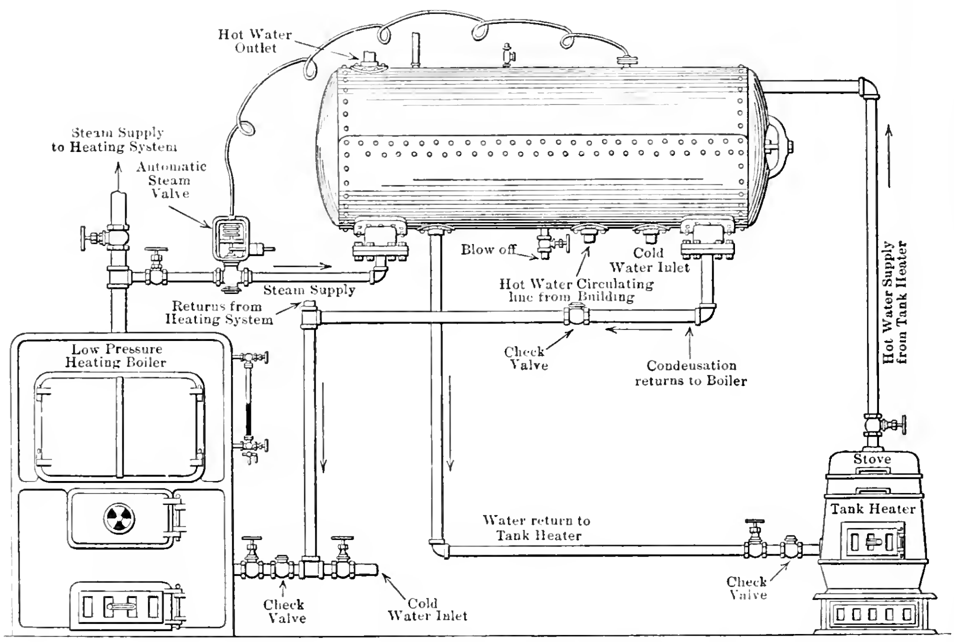


FIG. 7. DOUBLE SERVICE CONNECTION FOR PATTERSON HOT WATER TANK

The pipe coil heaters used by the *U. S. Government* are of three sizes fitted with copper coils or tubing, and are connected and supported as shown in Fig. 6.

TABLE 3

DIMENSIONS OF PIPE COIL HEATERS

See Fig. 6.

Tank	Coil	Connections			Tank Shell x Head
		Water	Steam	Heater	
18″ x 5′	15′ x 1¼″ O.D.	1″ x 1″ x ¾″	1¼″ x 1″	1¼″	3/16″ x ¼″
24″ x 7′	25′ x 1½″ O.D.	1¼″ x 1¼″ x 1″	1½″ x 1″	1½″	¼″ x 5/16″
36″ x 8′	35′ x 2″ O.D.	1½″ x 1½″ x 1″	2″ x 1½″	2″	¼″ x ⅜″

NOTE.—All tanks are equipped with a thermostatically controlled steam supply valve adjusted to close when the tank temperature is 140° F.

Tank and Heater Connections. The method of connecting a tank and coil for *double service* is shown in Fig. 7. By this arrangement it is possible to make use of steam during the heat-

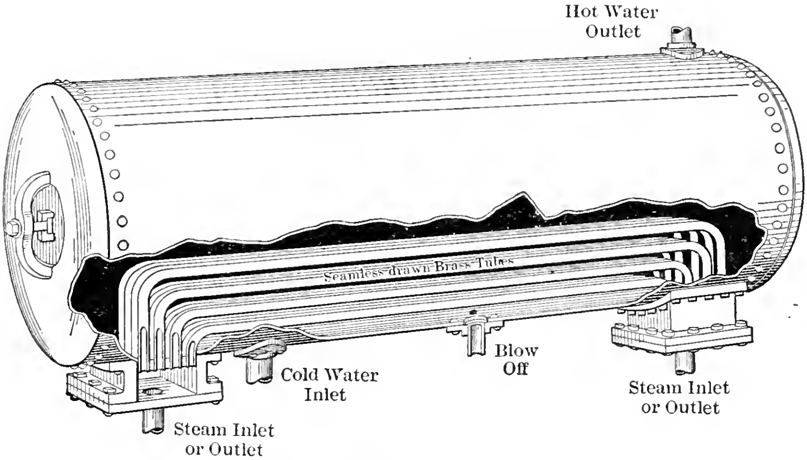


FIG. 8. PATTERSON HOT WATER TANK.

ing season, the condensation returning to the heating boiler by gravity, and during the balance of the year to employ an ordinary tank heater.

The steam supply (Fig. 7) may be under thermostatic control if desired, in which case a suitable check valve, trap or seal should be placed on the return, to prevent the condensation from backing up into the coils when the steam-valve is closed.

The *capacity* of the heating boiler must be large enough to *provide for the water heating requirements as well as the radiation supplied.* For example, each gallon of water heated from 50° F. to 180° F. is equivalent to, $1 \times 8\frac{1}{3} \times (180 - 50) / 250 = 4\frac{1}{3}$ sq. ft. of ordinary cast-iron steam radiation, and the rating of the heating boiler must be increased accordingly.

Where large quantities of water must be heated rapidly a continuous coil is not satisfactory, since the condensation formed in the first bends must flow through all those following, thereby reducing the efficiency. It is therefore customary to make use of short parallel tubes of seamless drawn brass or copper, so arranged as to provide for expansion and contraction. This arrangement is used in the *Patterson Hot Water Tank* (Fig. 8), and the capacities and principal dimensions are given in the table following.

TABLE 4

DIMENSIONS OF THE PATTERSON HOT WATER TANK

Number	Diameter	Length	Storage Capa'ty in Gals.	Min. Hourly Heating Capa'ty in Gals. from 50° F. to 180° F.	No. of Complete Tubes	Lineal Feet of 1¼" Tubing	Water Connections	Steam Connections		Approx. Weight
								Low Pressure	High Pressure	
1	24"	72"	150	225	6	32	2½"	1½"	¾"	800 lb.
2	24"	96"	190	285	6	38	2½"	1½"	¾"	950 "
3	30"	72"	210	315	8	43	3"	1½"	1"	1100 "
4	30"	96"	300	400	8	60	3"	2"	1"	1325 "
5	30"	120"	375	560	8	75	3"	2"	1"	1575 "
6	36"	96"	425	640	12	84	3½"	2"	1¼"	1875 "
7	36"	120"	530	800	12	110	3½"	2½"	1¼"	2200 "
8	36"	144"	640	960	12	134	3½"	2½"	1¼"	2500 "
9	42"	120"	720	1080	16	144	4"	3"	1½"	2675 "
10	42"	144"	860	1300	16	176	4"	3"	1½"	2975 "
11	48"	144"	1125	1690	20	225	4"	4"	1½"	3300 "
12	48"	168"	1300	1950	..	...	4"	4"	1½"	3900 "
13	48"	192"	1500	2250	..	...	5"	5"	2"	4300 "

The *Patterson* heater tanks are made of boiler plate not less than $^5/_{16}$" for shells and ⅜" for heads. Tanks 48" and 54" diam. have shells ⅜" thick and heads $^7/_{16}$" thick while the larger diameters are made of $^7/_{16}$" plate for shells, and ½" for heads. The tubes are seamless drawn brass, tested to 1000 lb. per sq. in. hydrostatic pressure, and 1¼" in diameter of No. 14 B. & S. gage. The tube heads are of heavy forged steel, and the assembled heater is tested to 200 lb. per sq. in. hydrostatic pressure.

Tank Regulators. There are a variety of *automatic thermostatic devices* for regulating the water temperature in the tank by controlling the steam supply. One of the simplest and most recent of these is shown in Fig. 9 and is equipped with a *Sylphon* bel-

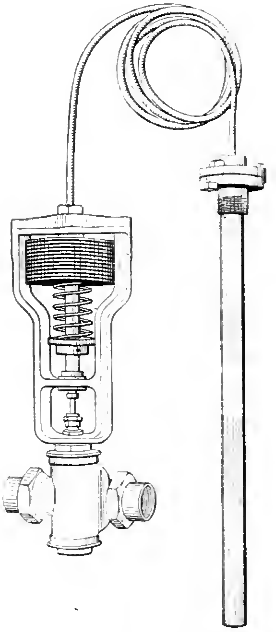


FIG. 9. IDEAL SYLPHON TANK REGULATOR WITH COMPRESSION SPRING.

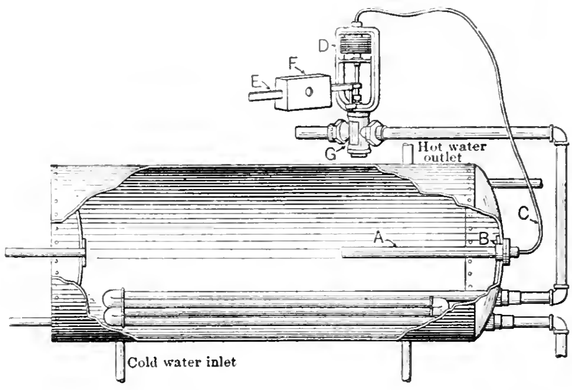


FIG. 10. IDEAL SYLPHON TANK REGULATOR ATTACHED TO STORAGE TANK.

lows and compression spring. The thermostatic element or brass bulb is filled with a volatile liquid, and is connected with the expanding bellows at the valve by means of an armored conduit. When the liquid in the bulb is vaporized the pressure created by this vapor will be trans-

mitted to the bellows which will expand and close the steam valve, against the compression spring, which normally keeps it open. These regulators are built in sizes from ½″ to 2½″ with a standard temperature range of 140° F. to 200° F. and will operate in any position, and are adapted to both stationary and portable service requirements.

The *application* of a tank regulator to a storage tank with continuous steam coil is shown in Fig. 10. This regulator operates exactly like the regulator shown in Fig. 9 except that a

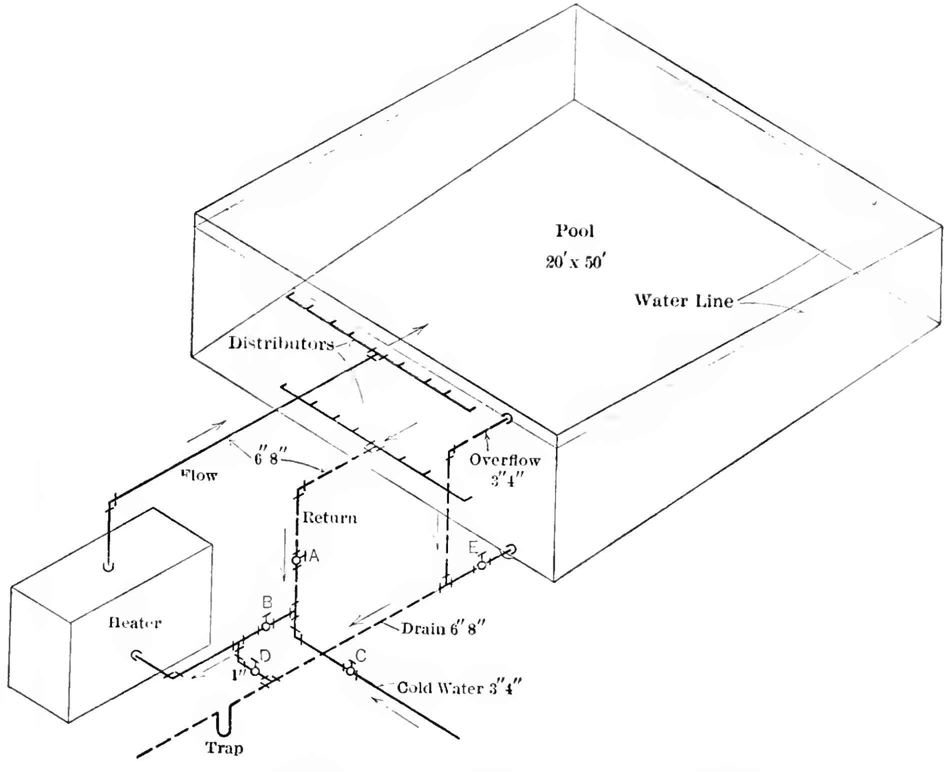


FIG. 11. SWIMMING POOL WITH WATER HEATER AND CONNECTIONS.

NOTE.—In operation, with tank full, valves *A* and *B* are open and *C*, *D* and *E* closed. To fill the tank, *through* the heater, close *A*, *E* and *D* and open *B* and *C*. The boiler should be placed as far below the pool as possible, in order to increase the motive head producing circulation through the heater.

weight is used to open the steam valve instead of a spring. Hence it can only be used for stationary service.

HEATING SWIMMING POOLS

Swimming pools may be heated: (1) by live steam discharged directly into the water in the pool using a suitable steam and water mixer, or (2) by steam coils placed in the pool itself, or (3) the water may be circulated through a water-heating boiler or a tank coil heater.

Steam and Boiler Horsepower Required. The steam required per hour is very easily calculated provided the volume of the pool and the temperature rise of the water are known. Thus, if we have 10,000 cu. ft. capacity in a certain pool with water entering at 50° F. to be heated to 75° F. in a period of 10 hrs., and the steam is at atmospheric pressure, or 212° F., the weight of

steam is, $S = [10{,}000 \times 62.5 \times (75 - 50)] / [(971.7 + 212 - 75) \times 10] = 1{,}333$ lb. per hour. No allowance has been made for heat losses through bottom and sides of tank to the earth which is at approximately 50° F. The boiler horsepower required is $[10{,}000 \times 62.5 \times (75 - 50)]/(33{,}524 \times 10) = 47$ b.h.p., and the B.t.u. loss to be supplied is 1,562,500 per hour.

Area of Steam Coils: If steam coils are to be used, the area of same for preceding example is readily found by reference to chart Fig. 5, as follows: Temperature difference between steam and average water temperature $= \left(212 - \frac{75 + 50}{2}\right) = 150°$; hence from the chart we may expect each sq. ft. of iron pipe coil to transmit 31,500 B.t.u. per hour, and therefore we will need about 1,562,500 / 31,500 = 50 sq. ft. or 100 lineal ft. of 1½″ pipe. At least 50 per cent should be added to the above amount of coil to allow for sluggish circulation in such a large body of water.

Size of Heating Boiler Required. If a water-heating boiler is to be used, the size of grate for the pool already considered is determined as follows:

$$G = \frac{Q}{H \times C \times E} = \frac{1{,}562{,}500}{12{,}000 \times 8 \times 0.70} = 23 \text{ sq. ft.}$$

G = grate area in sq. ft.
H = heat value of the coal, say 12,000 B.t.u. per lb.
C = rate of combustion, say 8 lb. per sq. ft. large boilers.
E = efficiency of boiler, say 70 per cent for this low water temperature.
Q = B.t.u. to be added to the water in the pool per hour.

Connections between Heater and Pool. When a water heater is to be used to operate directly on the water in the pool it is customary to connect it to the pool with suitable flow and return pipes so that the water may be circulated through the heater as often as may be necessary to obtain the desired rise in temperature. It is customary to allow a period of 10 hours for bringing the pool up to temperature when filling same with fresh water, and the piping connections should be so arranged as to send *the entering water through* the heater on its way to the pool (Fig. 11). After the tank has been filled the heater drafts should be so arranged as to maintain the required water temperature. This may be accomplished by the use of a tank regulator as already described for storage tanks. Since the motive head, producing circulation between pool and heater is very small, it is necessary to use large mains, and provide distributing pipes with numerous outlets on the tank end of the flow and return connections. In fact, all connections must be large, otherwise much time will be wasted in filling and emptying the tank.

The method of making the necessary connections between heater and tank, and the proper pipe sizes for a tank 20′ × 50′ in area with an average depth of 5′ are shown in Fig. 11.

CHAPTER VIII

DRAFT AND CHIMNEYS FOR HEATING BOILERS

DRAFT

Draft is the difference in pressure which causes the flue gases to rise in a chimney. If the gas inside a stack be heated each cubic foot of it will expand, hence its weight will be less than a cubic foot of colder outside air or gas; therefore the unit pressure at the base of the chimney, due to the column of heated gas, will be less than that due to a column of cold air or gas of the same height on the outside of the chimney.

This difference in pressure, like a difference in head of water causes a flow of cold air or gas into the base of the chimney. If, just at the point of entrance into the chimney, the cold incoming air is warmed up to the chimney temperature, the chimney will always be full of hot gas and the draft action will be continuous.

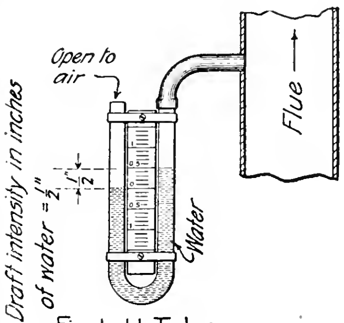


Fig. I. U-Tube Draft Gauge.

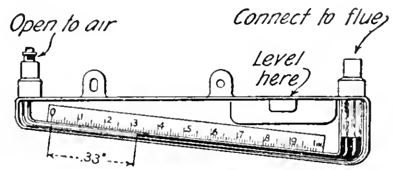


Fig. 2. Ellison Differential Draft Gauge
Draft intensity =.33" of water
This gauge reads to .01" of water direct.

Draft Measurement. The difference in pressure, or *intensity of draft* is usually measured in inches of water column as shown in Figs. 1 and 2.

As draft measurements are taken along the path of the gases, the readings grow less as the points at which they are taken are farther from the stack, until in the boiler ash-pit, with the ash-pit doors open for freely admitting the air, there is little or no perceptible rise in the water of the gage. The breeching, the boiler damper, the baffles, the tubes, and the coal on the grates all retard the passage of the gases, and the draft from the chimney is required to overcome the resistance offered by the various obstacles. The draft at the rear of the boiler-setting, where connection is made to the stack or flue, may be 0.5 inch, while in the furnace directly over the fire it may not be over, say, 0.15 inch, the difference being the draft required to overcome the resistance offered in forcing the gases through the tubes and around the baffling.

One of the most important factors to be considered in designing a stack is the pressure required to force the air for combustion through the bed of fuel on the grates. This pressure will vary with the nature of the fuel used, and in many instances will be a large percentage of the total draft. See chart of draft intensities, in chapter on "Boilers" under Section 2. In the case of natural draft, its measure is found directly by noting the draft in the furnace, for with properly designed ash-pit doors it is evident that the pressure under the grates will not differ sensibly from atmospheric pressure.

Theoretical Velocity of The Flue Gases. The velocity of hot gas rising in a chimney may

be calculated in the following manner (Fig. 3). The chimney is represented by column $C D$ with gas in same at temperature t_c and height h_o. Now h_c is the additional height of hot gas at temperature t_c which, when added to the column in the chimney, will just balance a column of air of equal cross-section at temperature t_o. In other words, the system $A\ B\ C\ D\ E$ is in equilibrium so long as column h_c is added to the hot gas column in the chimney. In practice, column h_c is lacking, hence the above system is unbalanced and flow occurs into the base of the chimney in virtue of the difference in head h_c. This is the head producing flow.

Assuming uniform cross-section throughout the system $A\ B\ C\ D\ E$, we have the following relation:

$$V_o : V_o + V_c = h_o : h_o + h_c = 460 + t_o : 460 + t_c$$

hence $h_o\,(460 + t_c) = h_o\,(460 + t_o) + h_c\,(460 + t_o)$ and $h_o\,(t_c - t_o) = h_c\,(460 + t_o)$

$$h_c = \frac{h_o\,(t_c - t_o)}{460 + t_o},$$

but h_c is the unbalanced head producing flow and creating the velocity. This velocity is, $V = \sqrt{2\,g\,h_c}$. Substituting, we have $V = 8.02\sqrt{\dfrac{h_o\,(t_c - t_o)}{460 + t_o}}$, where h_o is in feet, h_c is in feet of the hot gas at the temperature t_c, V is the velocity in feet per second, t_o and t_c are in degrees F., and $g = 32.16$.

It is found, in practice, that the above theoretical velocity is never obtained due to friction and other causes. The actual velocity will vary from 25 per cent to 50 per cent of the theoretical, depending on size of chimney and nature of interior surface. The smaller value may be used for flues in residences and the greater for large chimneys.

TABLE 1

DRAFT PRESSURES AND CORRESPONDING VELOCITIES*

Height of Water, Inches	Pressure in Lb. per Sq. Ft.	Velocity Ft. per Second	Velocity Ft. per Minute	Height of Water, Inches	Pressure in Lb. per Sq. Ft.	Velocity Ft. per Second	Velocity Ft. per Minute
h_w	P	V	$60\ V$	h_w	P	V	$60\ V$
0.1	0.521	21.2	1272	1.1	5.731	70.4	4224
0.2	1.042	30.1	1806	1.2	6.252	73.5	4410
0.3	1.563	36.8	2208	1.3	6.773	76.5	4590
0.4	2.084	42.5	2550	1.4	7.294	79.4	4764
0.5	2.605	47.4	2844	1.5	7.815	82.1	4926
0.6	3.126	51.9	3114	1.6	8.336	84.9	5094
0.7	3.647	56.1	3366	1.7	8.857	87.4	5244
0.8	4.168	59.9	3594	1.8	9.378	90.0	5406
0.9	4.689	63.6	3815	1.9	9.899	92.5	5550
1.0	5.210	67.0	4020	2.0	10.420	94.9	5694

Since the relation between the available heads or pressures may be expressed as follows:

$$^1/_{12}\,h_w\,K = h_c\,D \text{ also } P = {}^1/_{12}\,h_w\,K$$

then

$$h_c = \frac{h_w\,K}{12\ D}$$

and

$$h_w = \frac{12\,h_c\,D}{K},$$

in which K = density of water in the draft gage = 62.50 lb. per cu. ft.

D = density of flue gas or air at temperature t_c.

*NOTE.—Based on $V = \sqrt{2\,gh_c} = \sqrt{(2\,gh_w K) \div 12\,D}$, using D at 70° F. = 0.075. For other temperatures use corresponding value of D and see page 43.

h_w = height of water in inches in draft gage.
h_c = height of hot gas in feet at temperature t_c which causes flow, (Fig. 3).
V = velocity in feet per sec. of gas in chimney.
P = pressure in lb. per sq. ft. corresponding to h_w.

CHIMNEYS

Size of Chimneys Based on Velocity. *Wm. Kent* assumes a layer of gas 2″ in thickness as lining the chimney, and reducing its effective area by that amount. In this case the velocity as calculated above is assumed to be effective over the net area remaining within the inert ring of gas. See Fig. 4.

Intensity of draft *determines velocity* of flow through chimney *but the cross-sectional area* must be sufficient to pass the *necessary volume* of gas at this velocity, if the chimney is to have the proper *capacity*.

A method for calculating this volume was given under the chapter on "Fuels and Combustion" and a method for finding the cross-sectional area for an actual case is given in the following illustration:

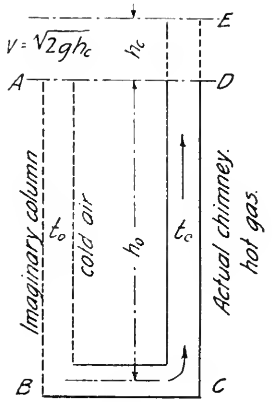


Fig. 3.

Example. Chimney calculation for heating boiler. (Based on velocity in flue.)

Given Data. Heat loss, including mains, etc., per hour = 1,500,000 B.t.u. to be supplied by boiler having a combined efficiency of 60 per cent. Calorific value of coal = 13,536 B.t.u. per 1 lb. and actual air required = 14.45 lb. per 1 lb. of coal or 144 per cent of the theoretical. Temperature at base of flue = 400° F., average in flue = 250° F., and outside air = 70° F. See chapter on "Fuels and Combustion." Height of chimney from grate to top = 50 ft. Actual velocity in flue is assumed ⅓ of the theoretical.

Required. Transverse area of chimney, and size of commercial flue lining to be used. See Table 3.

Calculations. Coal per hour $= \dfrac{1{,}500{,}000}{13{,}536 \times 0.60} = 184.7$ lb. Flue gas per hour $= 184.7 \times (14.45 + 1)/0.04618 = 61{,}900$ cu. ft., since chimney must pass the gas at 400° F. at which temperature density = 0.04618 lb.

Theoretical velocity, $V = 8.02\sqrt{h_o\dfrac{(t_c - t_o)}{460 + t_o}} = 8.02\sqrt{50\dfrac{(250 - 70)}{(460 + 70)}} = 33$ ft. per second, and hence actual velocity $= \frac{1}{3} \times 33 = 11$ ft. per second.

Area $= A = \dfrac{Q}{V} = \dfrac{61{,}900}{11 \times 3{,}600} \times 144 = 225$ sq. in. An 18 x 18 in. rectangular fire clay flue lining is 15½ x 15½ in. inside or 240 sq. in. in area, and an 18 in. inside diameter round lining is 254.5 sq. in. in area.

Either lining will be satisfactory; the latter is preferable as a flue, but the rectangular lining is more readily enclosed in the masonry.

Size of Chimney Based on Head Available and Frictional Resistances. Since, for every kind of fuel and rate of combustion there is a certain draft with which the best general results may be obtained, it is generally customary to base the design of large stacks upon the frictional losses to be overcome throughout the system, including grate, fuel bed, boiler setting, and breeching losses. Such a method of designing chimneys or stacks is considered under "Stacks for Power Boilers" in Section 2.

Chimneys for Heating Boilers. The proper size of chimney for a heating boiler of given capacity may be calculated on the basis of air required for combustion and actual flue velocities or

frictional resistances to be overcome as already stated above, or it may be taken from tabulated data obtained by a similar method of calculation in which a reasonable allowance has been made for friction or reduced velocity. It must be constantly kept in mind that the friction varies as the extent or area of the rubbing surface, hence for the same length of flue of given cross-sectional area the section having the least perimeter will offer the least friction to the flue gases and therefore require the least draft, or give the highest flue velocity.

Selection of Chimney Flues. The selection of chimney flues for heating boilers must depend upon the judgment of the heating engineer, but it is believed that the table herewith, by *Prof. R. C. Carpenter*, will very much assist the engineer in selecting flues.

It is necessary that *area* and *height, thickness of walls, general structure*, and the *position of the top outlet* with reference to the building and other buildings near by should be carefully noted and observed in the selecting or building of a flue.

The figures given under the varying heights of chimneys are diameter measurements in inches, or, the side of a square—the theory being that the spirally ascending column of smoke and gases will make a 12″ x 12″ flue no more effective in practical working area than a twelve-inch round flue. Rectangular shapes may be used if the area is equal and the difference in width and breadth are not extreme. A maximum ratio of 2 : 1 for the internal dimensions should not be exceeded. No flue should be less than 6″ x 6″ inside, and better, 8″ x 8″.

TABLE 2

CHIMNEYS FOR STEAM AND HOT WATER BOILERS

Direct Radiation*		Height of Chimney Flue				
Steam in Square Feet	Water in Square Feet	30 Ft.	40 Ft.	50 Ft.	60 Ft.	80 Ft.
250	375	7.0	6.7	6.4	6.2	6.0
500	750	9.2	8.8	8.2	8.0	7.6
750	1150	10.8	10.2	9.6	9.3	8.8
1000	1500	12.0	11.4	10.8	10.5	10.0
1500	2250	14.4	13.4	12.8	12.4	11.5
2000	3000	16.3	15.2	14.5	14.0	13.2
3000	4500	18.5	18.2	17.2	16.6	15.8
4000	6000	22.2	20.8	19.6	19.0	17.8
5000	7500	24.6	23.0	21.6	21.0	19.4
6000	9000	26.8	25.0	23.4	22.8	21.2
7000	10500	28.8	27.0	25.5	24.4	23.0
8000	12000	30.6	28.6	26.8	26.0	24.2
9000	13500	32.4	30.4	28.4	27.4	25.6
10000	15000	34.0	32.0	30.0	28.6	27.0

* When a considerable amount of *indirect* radiation is to be used increased boiler capacity is necessary, and in many cases such demands require a larger chimney flue for the same number of square feet of radiation used. (See also "Equivalent Ratings" in the chapter on "Heating Boilers.")

It will be noted that the size or area of a chimney is a function of the height and can be made less as height or flue temperature increases. The data in this table have been plotted in the form of curves (Fig. 4). Height of chimney is from grate level to top of flue. A table of *chimneys for power boilers* by *Wm. Kent* is also given, and can be used in connection with the above table on the basis that 1 h.p. = 100 sq. ft. of direct steam radiation. See "Stacks for Power Boilers" in Section 2.

A simple check rule for chimneys for steel heating-boilers as used by the *U. S. Treasury Dept.* is based on, H = height of stack in feet, A = area of grate in sq. ft., S = area of stack in sq. ft. (1) $S = \frac{A}{\sqrt{H}}$ for anthracite and bituminous lump coal, oil and gas. (2) $S = 1.25 \frac{A}{\sqrt{H}}$ for small sizes of anthracite and bituminous coals. Chimneys for cast-iron boilers are calcu-

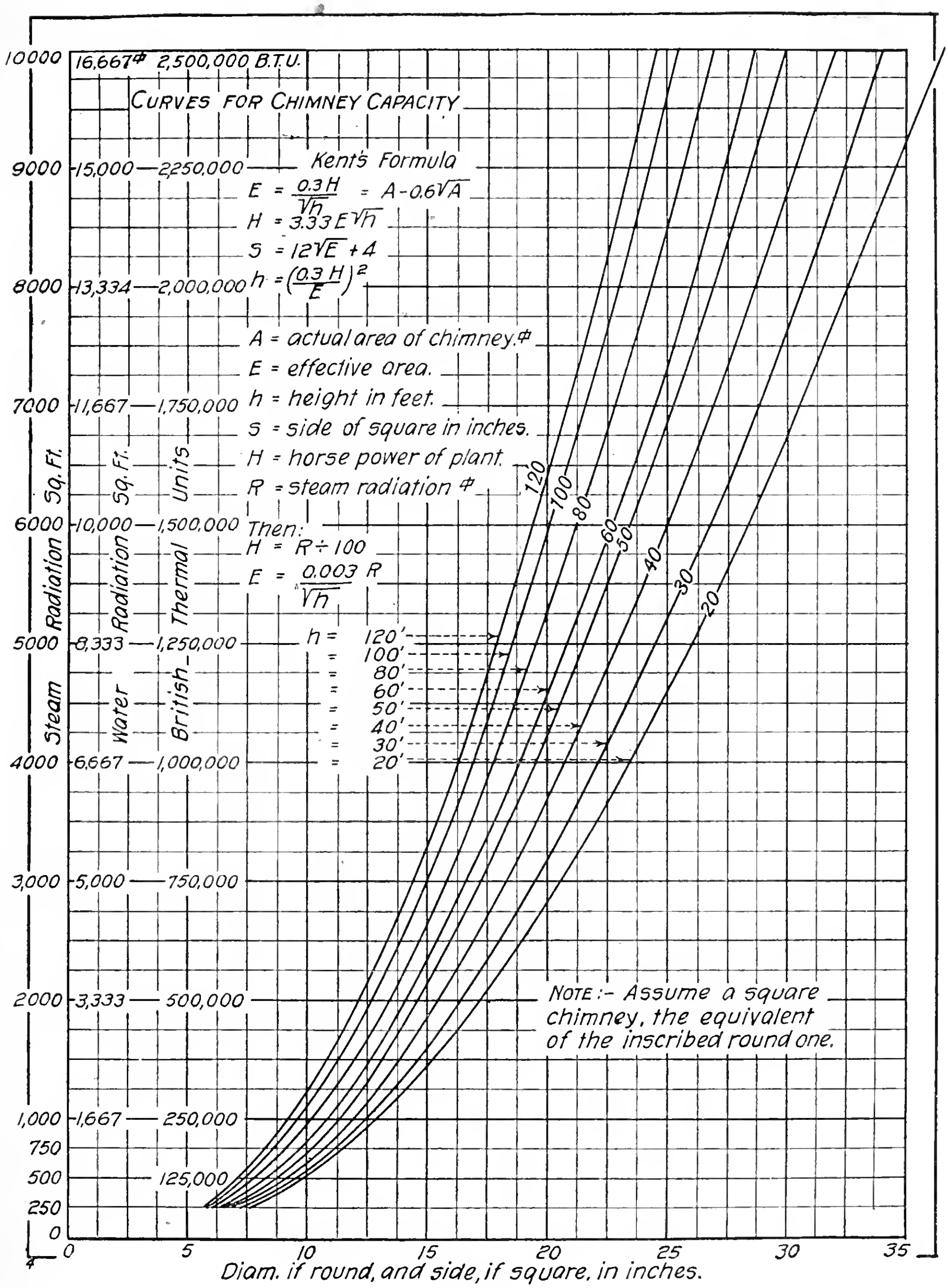

10000 16,667# 2,500,000 B.T.U.
CURVES FOR CHIMNEY CAPACITY
9000 15,000 2,250,000
Kent's Formula
E = 0.3H/√h = A-0.6√A
H = 3.33E√h
S = 12√E + 4
h = (0.3H/E)²
8000 13,334 2,000,000
A = actual area of chimney.#
E = effective area.
h = height in feet.
7000 11,667 1,750,000
S = side of square in inches.
H = horse power of plant.
R = steam radiation #
Then:
H = R ÷ 100
E = 0.003 R/√h
6000 10,000 1,500,000
Steam Radiation Sq. Ft.
Water Radiation Sq. Ft.
British Thermal Units
5000 8,333 1,250,000
h = 120' 100' 80' 60' 50' 40' 30' 20'
4000 6,667 1,000,000
3,000 5,000 750,000
2000 3,333 500,000
1,000 1,667 250,000
750
500 125,000
250
0
120 100 80 60 50 40 30 20
NOTE:- Assume a square chimney, the equivalent of the inscribed round one.
0 5 10 15 20 25 30 35
Diam. if round, and side, if square, in inches.

FIG. 4.

lated by the following formula: (3) $S = 0.75 \frac{A}{\sqrt{H}}$ Another check rule for chimneys for heating boilers is to take $^1/_{10}$ the radiating surface in sq. ft. of direct steam radiation as the area of the flue in sq. in. This is for hard coal; for soft coal add 20 per cent.

Chimney Construction. The following *details of construction* must be carefully observed, in addition to making the chimney of the proper size, or failure may result.

The value of the flue depends on *area* and *velocity*. Velocity alone is no proof of good draft—there must be also sufficient area to carry the gases.

The chimney-top should run above the highest part of the roof and should not be less in height than shown in Table 2, as chimneys under 30 feet in height are unreliable in their action. All chimneys should clear the ridge by at least 2 feet (Fig. 5).

The *American Radiator Co.'s* Institute of Thermal Research states the following concerning height of flue:

" Our experience is that any flue under 40 feet in height will give an erratic draft. Some days the draft will be good, and other days it will be poor; and this variation is not always due

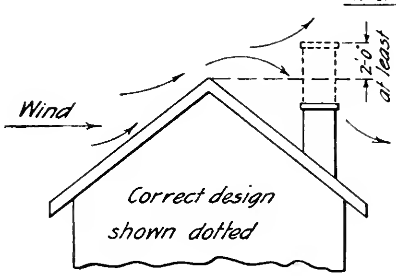


Fig. 5.

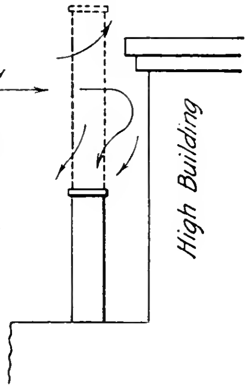


Fig. 6.

to the direction or force of the wind, but is more often due to the quality or fineness of the fuel. In burning soft coal, the quality known as 'run of mine' will fill up the air space and an intense draft is required to give the fuel sufficient air. The same is true of 'caking' soft coal, and of the sizes of hard coal known as 'pea' and 'buckwheat.'

" *L. Ser*, a French physicist, says in regard to heights: 'A height of ten meters (about 40 feet) is insufficient for a regular draft.'

" *Rietschel* does not advise a chimney less than 16 meters (52½ feet) in height. As a result of *Rietschel's* work and advice along these lines, the chimney drafts in Germany are even better than they are in France, and, as a result, they are able to use boilers in those countries which would be an entire failure with us."

The chimney should be so located with reference to any higher buildings nearby that wind-currents will not form eddies and force the air downward in the shaft as shown in Fig. 6.

The flue should run as nearly straight as possible from the base to the top outlet. The outlet must not be capped so that its area is less than the area of the flue. The flue should have no other openings into it but the boiler smoke-pipe. Sharp bends and offsets in the flue will often reduce the area and choke the draft, and the flue must be free of any feature which prevents full area for the passage of smoke.

If the flue is made of tile, the joints must be well cemented, or all space between the tile and brick-work filled in tightly. There must be no open crevices into the flue where the tile sections meet, otherwise the draft will be checked.

If the flue is made of brick, the stack should have outside walls at least eight inches thick to insure safety. The inside joints should be well struck, and each course should be well bedded and free from surplus mortar at the joints. The exposed bricks at the top of a brick chimney should be laid in cement mortar to prevent the acid fumes and rain from cutting out the joints as will happen if lime mortar is used. The most desirable location for a chimney is near the center of the building, as all walls are then kept warm. Chimneys on outside walls may fail, due to the chilling effect of cold walls.

If there is a soot-pocket in the flue below the smoke-pipe opening, the clean-out door should always be tightly closed. If this soot-pocket has other openings into it from fireplaces or other connections, these openings check the draft and prevent best results.

The smoke-pipe should not extend into the flue beyond the inside surface of the flue, otherwise the end of the pipe cuts down the area of the flue.

The joints, where the smoke-pipe fits the smoke-hood of the boiler, or where the pipe enters the chimney, should be made tight with boiler putty or asbestos cement. Foundations must be ample to prevent settlement and cracks which will allow cold air to enter.

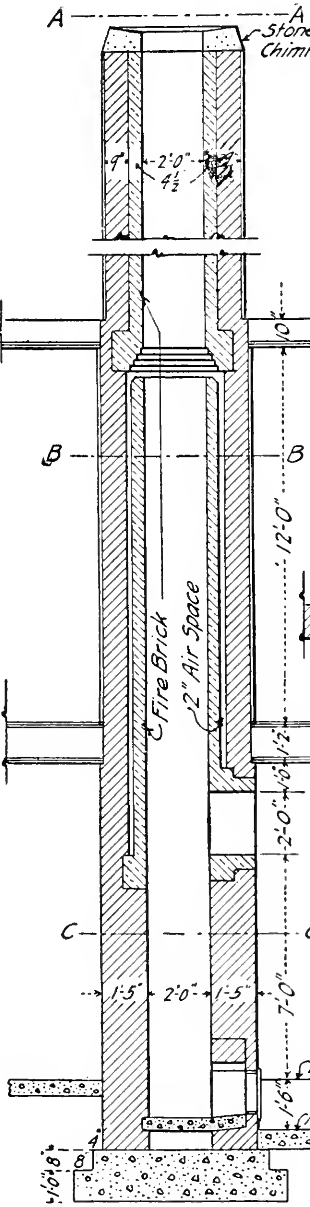
TABLE 3

FIRE CLAY FLUE LININGS

(*Robinson Clay Product Co.*, Akron, Ohio.)

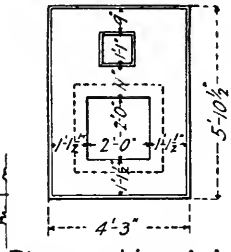
Number	Nominal Size, Inches	Actual Outside, Inches	Actual Inside, Inches	Weight per Ft., Lb.
		RECTANGULAR		
1	4½ x 8½	4¾ x 8⅝	3¼ x 7	16
2	4½ x 13	4¾ x 13¼	3 1/16 x 11¾	20
3	4½ x 18	4½ x 17	3⅞ x 15½	28
4	6 x 12	6 x 12	4½ x 10½	22
5	7 x 7	7¼ x 7¼	5¾ x 5¾	15
6	8½ x 8½	8½ x 8½	7¼ x 7¼	20
7	8½ x 13	8½ x 13	6⅞ x 11⅝	29
8	8½ x 18	8½ x 18	6½ x 16	37
9	13 x 13	13 x 13	11¼ x 11¼	42
10	13 x 18	13 x 18	10¾ x 15¾	58
11	18 x 18	18 x 18	15½ x 15½	74
		ROUND		
1	6	7½	See Nominal Size	15
2	7	8½		16
3	8	9		22
4	9	10½		26
5	10	12		30
6	12	14		45
7	15	17⅛		60
8	18	20⅞		80
9	20	23		90
9–a	21*	....		100
9–b	22*	....		115
10	24	27		130
10–a	27*	....		170
11	30	35		230
11–a	33*	....		245
11–b	36*	....		290

NOTE.—Round lining is rated by inside dimensions. Rectangular lining is rated by outside dimensions.
Round lining may be also obtained in the following diameters, 4, 5, 21, 22, 27, 33 and 36 inches, and rectangular lining 6″ x 16″, and 14″ x 16″.
* *Arthur N. Pierson & Co.*, New York.



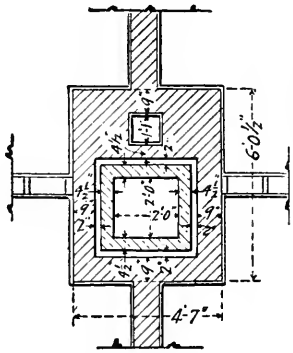
Section thro Chimney - W.L.Conley

FIG. 7.



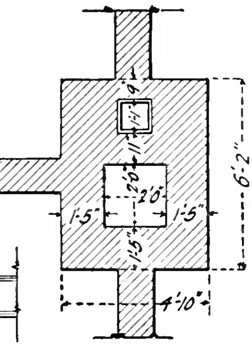
Plan on Line A-A

FIG. 8.



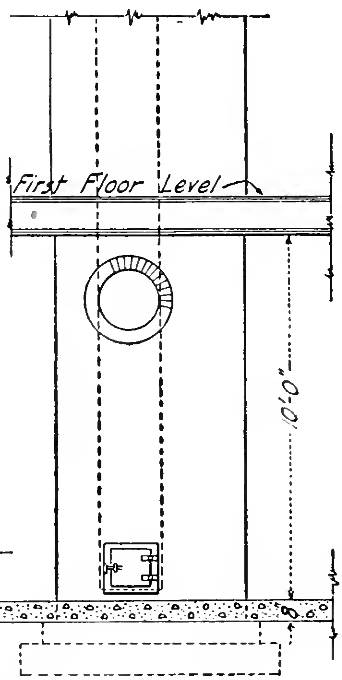
First Story Plan on Line B-B

FIG. 9.



Basement Plan on Line C-C.

FIG. 10.



Elevation of Chimney

FIG. 11.

Scale ¼"=1'

Brick Chimneys and Linings. Brick chimneys of large size may be lined in whole or in part with fire-brick as shown in Figs. 7 to 11 which also show all details of construction.

Fire clay flue linings are used in the best practice for small and medium size of flues giving

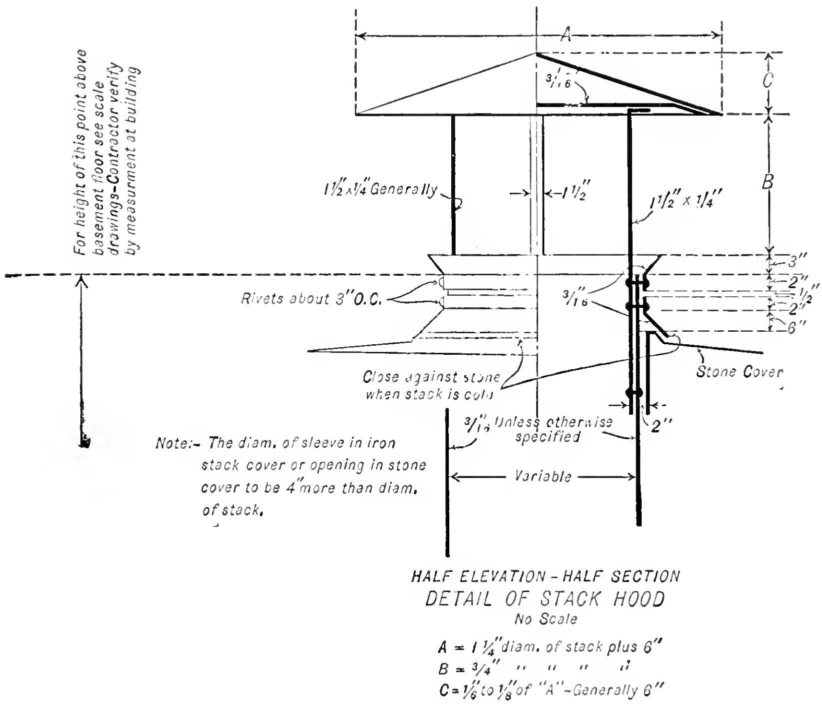


FIG. 12. STEEL CHIMNEY TOP.

a smooth continuous channel for the passage of the gases. The commercial sizes, weights, etc., will be found in Table 3.

Other linings of cast iron or steel are often used, the latter for large chimneys especially, and in many cases these metal linings are placed in a larger brick flue, so that the area between the brick flue and metal stack may be used as a vent space. The following specification for a 90 ft. steel stack enclosed as stated above will indicate the method of construction in this case:

Specification for Steel Smokestack Enclosed in Brick Chimney. (*U. S. Treasury Dept.*) Smokestack to be of height shown on plans and 44 inches internal diameter. The first 45 feet to be constructed of $\frac{3}{8}$-inch tank steel, and the remainder $\frac{1}{4}$-inch thick, all in sections about 6 feet long. All vertical seams of sections are to be lap-riveted, while the circular seams are to be made with outside butt straps 4 inches wide and the same thickness as the sections to which they are riveted. The rivets on the vertical seams are to be $\frac{1}{2}$-inch in diameter and spaced 4 inches on centers. The circular butts are to be riveted to top edge of each section of stack by $\frac{1}{2}$-inch rivets.

The stack is to have tight fitting cast-iron clean-out doors not less than 24 inches by 30 inches near the bottom, with hinges. The opening for doors must be re-enforced suitably by a frame

of 1½-inch by ½-inch bar iron and fitted with a substantial cast-iron door frame properly riveted on. The bottom of stack is to be re-enforced by steel collar 3 inches by ⅝-inch riveted to stack by ½-inch rivets. The top of stack is to be re-enforced with $^3/_{16}$-inch flaring piece, riv-

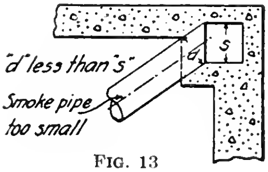


FIG. 13

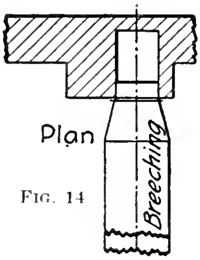


FIG. 14

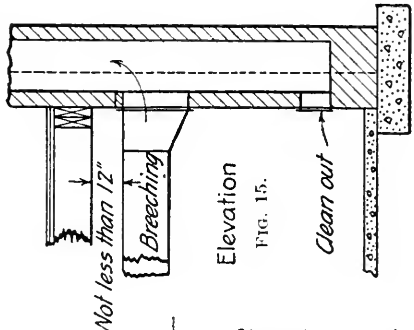


FIG. 15.

See FIGS. 14 and 15 for method of connecting a full sized breeching.

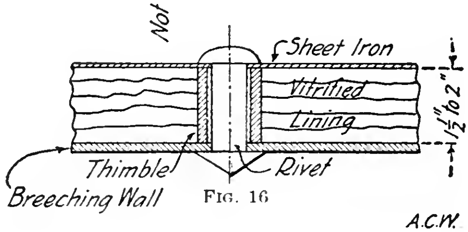


FIG. 16

FIGS. 13 TO 16.

eted to stack by ½-inch rivets. A $^3/_{16}$-inch tank steel flashing, covering the opening through chimney, will also be provided.

The top of stack is to be further provided with a $^3/_{16}$-inch double steel plate cover supported by four 1½-inch by ¼-inch bar iron supports riveted to stack and bottom plate of cover, as shown in detail drawing (Fig. 12).

Stack must be properly braced to shaft by 1½-inch by ¼-inch bar iron bolted loosely around stack and let into brickwork of shaft at intervals not exceeding 12 feet. Each pair of rods to be at right angles to those next adjacent above and below. The stack to be provided with cast-iron bedplate 50 by 50 inches and 2 inches thick, with necessary ribs. Smoke stack to be painted with one coat of best quality graphite paint, both inside and out.

Breechings for Heating Boilers. The *horizontal smoke connection* between boiler and chimney is often referred to as the *breeching.* Strictly speaking this term applies only to the actual opening into the chimney or stack. This connection should be about 10 per cent greater in area than the chimney, and should ordinarily never exceed in length 50 per cent of the height of chimney. The extra size is allowed for accumulations of soot and ash in the horizontal run. Suitable cleanouts should be provided at elbows for access to the interior.

The connection to the chimney should be made full size of breeching and Figs. 13, 14 and 15 show typical conditions to be met with in practice in making these connections.

The *round breeching* is preferable for one or two boilers, unless headroom or other reasons require a rectangular shape. Many designers make round breechings the same area as stack, but add 2 inches in depth to *rectangular breechings* as indicated above, and, where possible, rectangular breechings are made the same width as stack.

The *insulation of breechings* is necessary to avoid excessive heat loss, and either an *external covering* of asbestos paper or plastic asbestos may be used, or an *internal lining* of some suitable vitrified material or brick may be used. The plastic material must be properly secured on ex-

panded metal when applied to the exterior, and requires great skill in its application. If a vitrified lining is used, such as the *Johns-Manville* "Vitribestos", it must be securely fastened to the interior by means of bolts or rivets and washers, or by a thin sheet-iron cover-plate as shown in Fig. 16.

Smoke connections or breechings must not come within 12 inches of the under side of wooden beams or floors. The *smoke connection on the boiler* may or may not be of the same size as breeching to chimney. Each manufacturer determines this size to suit his boiler, and assumes average conditions will exist, so that for many special conditions which constantly arise larger flue connections may be necessary. This simply means that the gases pass through the smaller connection on the boiler at increased velocity provided the chimney flue is capable of developing the proper draft intensity.

CHAPTER IX

DIRECT STEAM HEATING

Systems in Use. Systems for heating with direct steam radiators are broadly divided into two general classes, known as: (1) *gravity circulating* systems, and (2) *mechanical circulating* systems. The distinguishing characteristic is the manner in which the water of condensation from the radiators is returned to the boiler.

In the first type the condensate enters the boiler by gravity, due entirely to the *static head* existing in the returns, and the system is a closed circuit. The steam pressure existing in the boiler, mains, and radiators is the same, except for friction pressure losses due to the flow of steam to the heating surfaces (Figs. 1, 4 and 7).

In the second type the condensate is allowed to return to a receiver or feed water heater and is then forced into the boiler by a *pump* or *return traps* or both. This is not a closed system, and the pressure in the boiler may be much higher than that in the mains and radiators (Fig. 0). The receiver is usually vented to the atmosphere, and in the case of vacuum systems an additional pump is attached directly to the returns and arranged to discharge the condensation into the receiver or heater. See Figs. 7, 8 and 9, under the chapter on "Exhaust Steam Heating," and "Mechanical Vacuum Systems" as described in this chapter.

GRAVITY STEAM HEATING SYSTEMS

Gravity circulating systems are further divided into *one* and *two* pipe systems with basement mains supplying risers to the various floors above (Figs. 1, 4 and 7) or with overhead mains supplying drop risers to the floors below. In the latter system the steam and water of condensation in the risers flow in the same direction, so that less friction is produced as counter currents do not occur and smaller pipe sizes may be used. The overhead system is very commonly spoken of as the *Mill's* system (Fig. 0).

One-Pipe Gravity Systems. The *one-pipe circuit system* with basement mains is probably the simplest, and most common gravity system in use. (Figs. 1 and 4.) The steam main rises close to the basement ceiling just above the boiler, and then grades down uniformly from this high point with a fall of 1″ or ¾″ in 10′–0″. When the last radiator has been served the main drops below the boiler water line, and its size is reduced, as on the run back to the boiler it carries only condensation and is known as a *wet return*. This return may be run above the boiler water line if necessary, and is then called a *dry return*. Return mains are graded 1″ in 30′–0″ in gravity work. In either case an automatic air valve must be installed on the end of the main at the drop (Fig. 2) to vent the same when air collects in the piping.

The elevation of the end of the steam main with respect to the boiler water line must be carefully determined, in order that water may not back up from the boiler and flood the main, including the air valve and branches. For a complete discussion of this point see "Piping Sizes" in this chapter. It is customary to *maintain at least* 12″, *and preferably* 18″, between the under side of main at the drop and the normal water line of the boiler to provide for contingencies.

In operation it will be noted that steam and water flow in the same direction through the one-pipe steam main, and in opposite directions through the basement branches, risers, and radiator branches. This necessitates larger piping and valves than in any other steam system,

and especially is this true of the main, which must be run full size from boiler to drop, unless dripped as shown under piping details. See Fig. 6, and elevation of drip connections in Fig. 53.

The *one-pipe relief system* (Figs. 4, 6 and 8) is very similar to the one-pipe circuit system except that the risers are dripped individually into the return, and the steam main carries no ra-

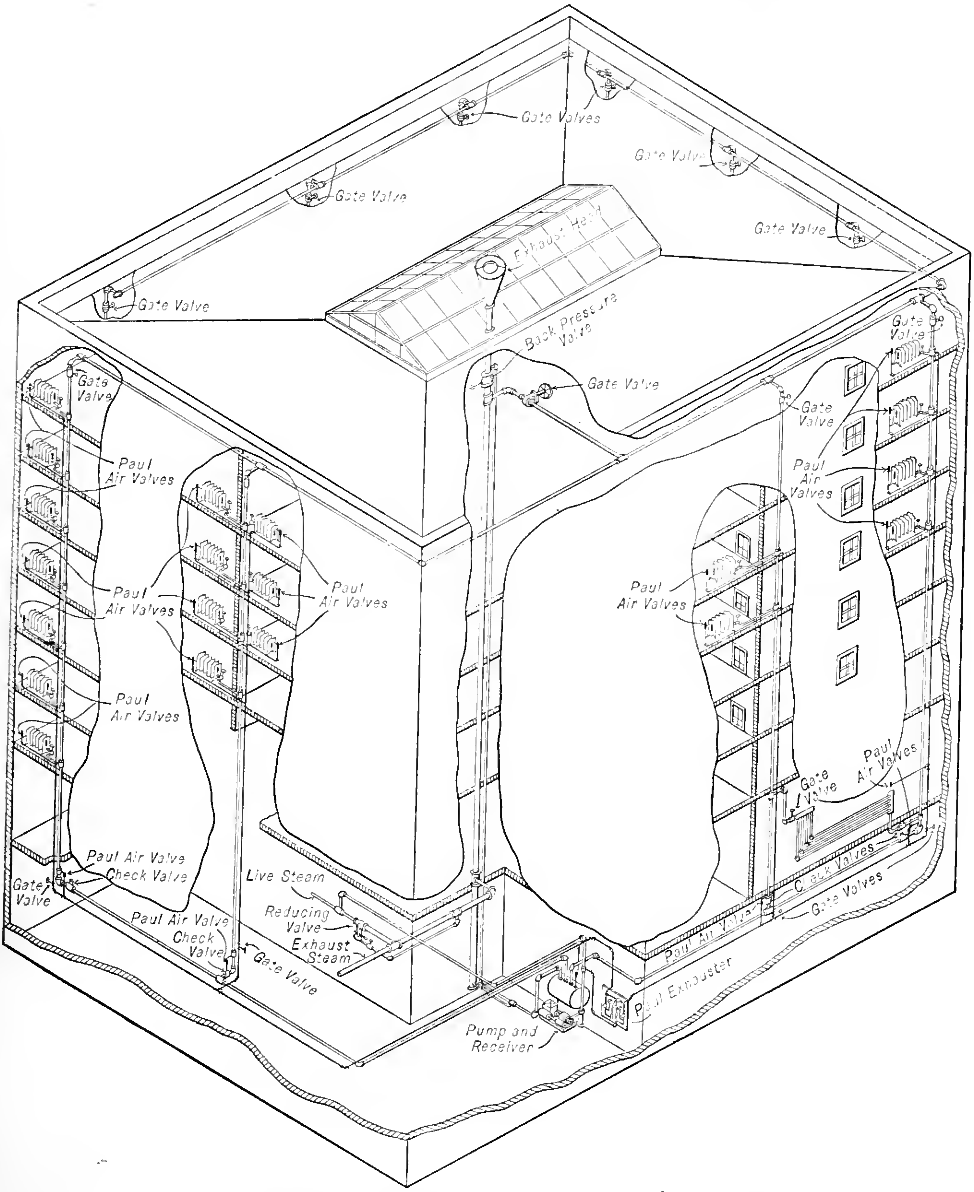
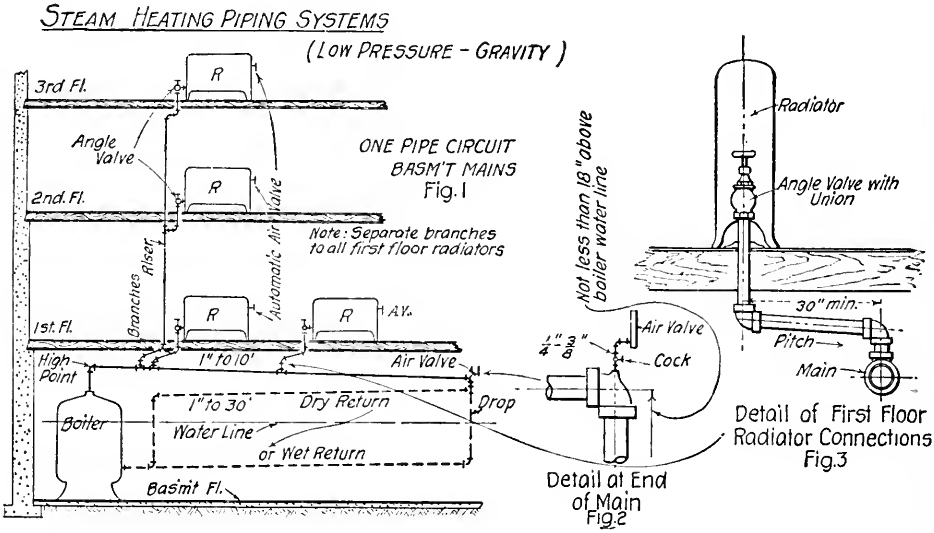


FIG. 0. SINGLE PIPE OVERHEAD DOWN-FEED OR MILL'S SYSTEM.
(Paul Air Line)

diator condensation, and is itself dripped at intervals into the return main, which may run *dry* or *wet*. This makes it possible (1) to reduce the size of main as radiation is taken off, (2) to use smaller branches, and (3) to run the main much closer to the basement ceiling, a very important consideration where basement space is valuable.

A combination of the one-pipe relief and the two-pipe system is sometimes used in large installations, the latter being used for the first and second floors, and the former for the upper floors of high buildings. In this way the amount of condensation flowing down the one-pipe risers against the steam is much reduced and smaller risers may be used.

The application of the one-pipe system with gravity circulation and *basement mains* to buildings of 20 stories in height, is not at all unusual, and if the piping is properly designed for a friction pressure loss of 1 oz. per 100 ft. of run the circulation of steam and the return of the water of condensation will be found entirely satisfactory. The use of the two-pipe gravity system or a vacuum system in a job of this size is often considered the only alternative, but recent installations in New York city have entirely justified the simpler system.

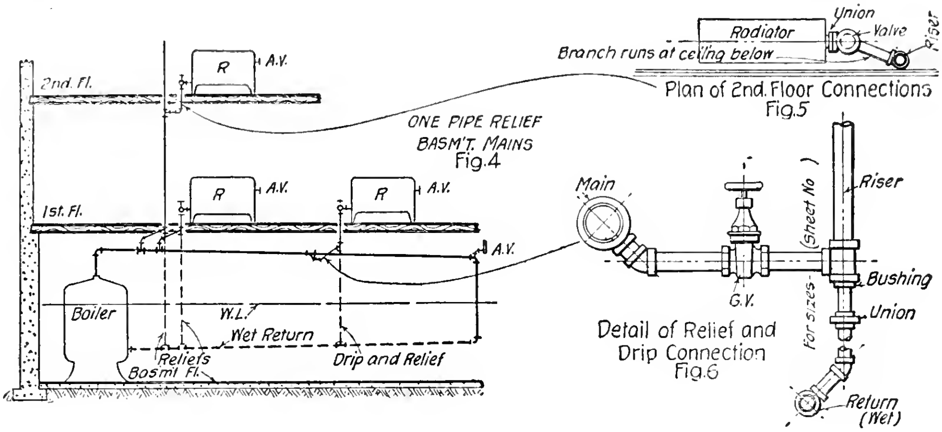


In the case of long narrow buildings heated by a gravity system it may be necessary to provide a deep boiler pit so that the elevation of water in the return connections will not flood the far end of the steam mains. (Fig. 58.)

Two-Pipe Gravity Systems. The *two-pipe system* with basement mains (Fig. 7) is often used in large buildings, and in *all work where indirect radiation is installed.* This system can be readily adapted to mechanical vacuum systems, and is very extensively used in this connection. It will be noted, however, that when applied to a gravity system the return from each radiator is *separately sealed,* either by dropping below the water line to a wet return or else by using drip loops, as shown at the left in Fig. 7, before connecting to a dry return. Even in one-pipe work all drips or reliefs are sealed as shown in Fig. 8. If this precaution is not taken steam may enter a drip or return from the outlet end and cause knocking in the system due to counter currents of steam and water of condensation.

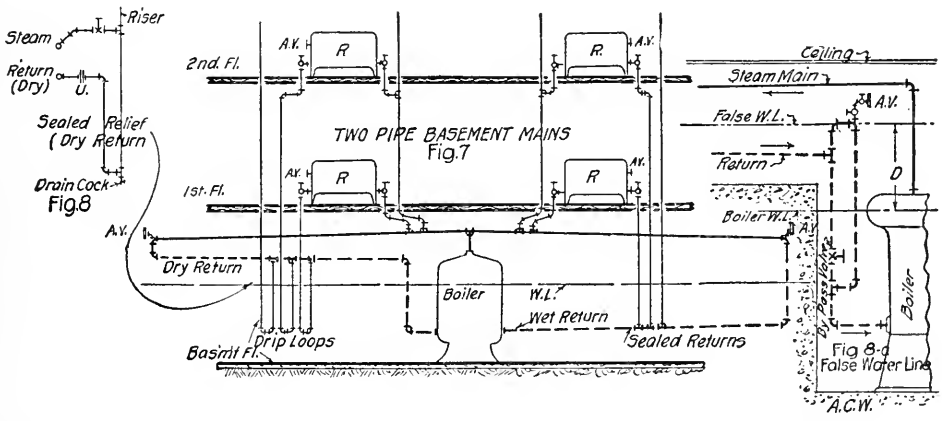
It is an invariable rule in steam heating practice that any drip, relief, return riser or connection from the steam to the return side of the system *must be sealed.* This may be done by connecting below the water line, or else by using a running trap or a return trap somewhere on this connecting line.

Automatic Radiator Air Valves for Gravity Systems. The *automatic removal of air* from steam radiators must be provided for if the highest efficiency of the radiating surfaces is to be realized in gravity circulating systems. Manually controlled air valves or cocks are usually neglected, and are seldom used for steam radiators although their use is quite general for hot-



water radiators, for which latter service there are very few satisfactory valves of the automatic type. See the chapter on "Hot Water Heating."

The proper *location* of the air valve on steam radiators is at the end of the radiator opposite the steam inlet, and as near the bottom of the radiator as possible, since air is heavier than steam at the same temperature. In practice, however, the manufacturer of radiators usually places



the air valve tapping about two-thirds the height of the radiator from the floor in order to prevent possible flooding of the valve.

The *expansion post* type of air valve (Fig. 9) made by the *Monash-Younker Co.* for steam radiators is so arranged that when steam strikes the expansion post *F* it expands, and raises the hollow float *C* until the valve point at top of float engages the adjustable seat E, and thereby prevents the escape of steam. Should water enter the valve the float operates to close the port and prevent its escape. In case neither steam nor water are present air may pass in or out through

the discharge port in E, the air tube D tending to facilitate its removal, and the four-way drain helping to keep the valve clear of condensation.

These valves are usually provided with a ⅛" male thread connection, and the air valve boss on the radiator is tapped accordingly. In government and institutional work, as well as in schools, the use of air valve *holders* is often resorted to in order to keep the valves in place, and in the proper vertical position. In this case it is necessary to cast a second boss 3¼" above the regular tapping on the end section of the radiator to provide for the air valve holder, as shown in Figs. 10 and 11. These special tappings and holders must be definitely called for in the specifications if desired.

The *expansion chamber* type of air valve (Figs. 12 and 13), known as the *Norwall Siphon*, depends upon the contraction and expansion of air which is trapped by a suitable water seal in an annular air chamber. The valve is constructed entirely of metal, and made by the *American Radiator Co.*

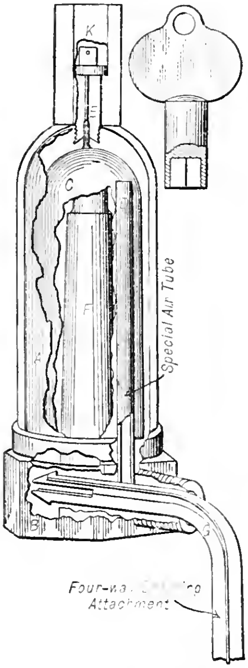


FIG. 9. SECTION OF AUTOMATIC STEAM-AIR VALVE.

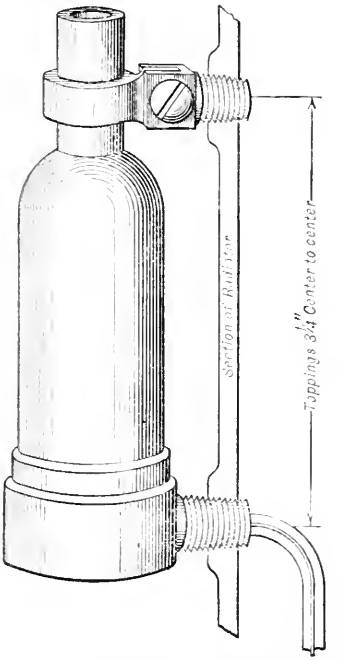


FIG. 10. EXTERIOR OF STEAM-AIR VALVE WITH HOLDER.

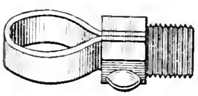


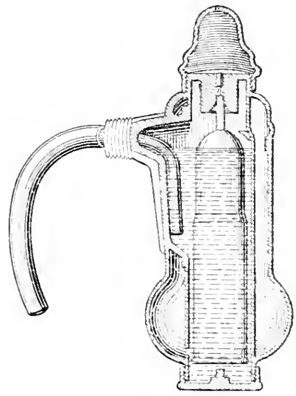
FIG. 11. AIR VALVE HOLDER.

A sealed metal float is contained in the inner or float chamber, which is connected to an outer or air chamber by a small hole at the bottom. Water of condensation is retained in the well formed by the location of the radiator connection at the top of the valve.

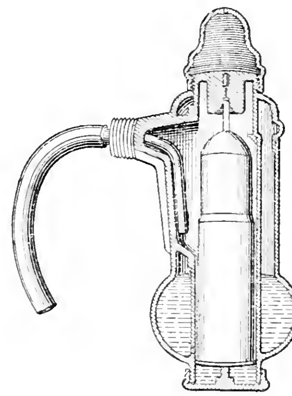
When steam reaches the valve the air in the outer chamber expands, forcing the water into the inner chamber, raising the float and closing the valve; when the valve cools this air contracts, drawing the water from the float chamber into the air chamber. The float then drops and opens the valve. This action is practically instantaneous.

On "water-logged" radiators the valve closes tight against the escape of water, but the instant the water in the radiators falls away, the siphon automatically discharges the surplus

water in the valve back into the radiator, and the valve recommences venting. Should dirt interfere with the operation of the valve, the bottom cap is so fitted that it can be easily removed and the valve cleaned.



Valve Closed



Valve Open

FIG. 12. NORWALL SIPHON STEAM-AIR VALVE. FIG. 13.

SPECIAL GRAVITY SYSTEMS

In addition to the low pressure gravity systems already described there are a large number of special steam heating systems known as *vapor*, *vacuum* and *air line* systems, also operating with gravity return of the water of condensation.

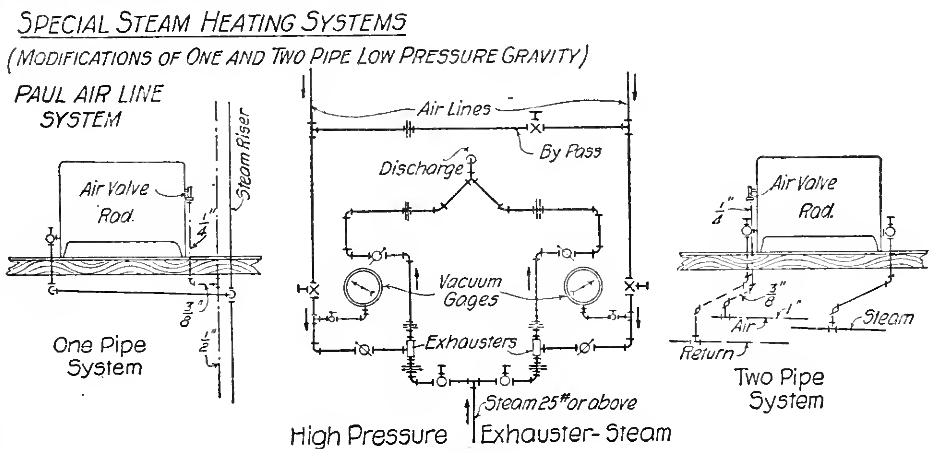


FIG. 14. FIG. 15. FIG. 16.

PAUL SYSTEM.—Steam and return piping laid out as for one-pipe gravity return. Exhauster for less than 2,500 pounds is a water-driven vacuum pump, pressure at least 20 pounds per square inch. Larger systems use a high-pressure steam jet (see above), or if steam is not available use motor-driven vacuum pump, about ¼ H.P. Use 1 inch air mains in basement and gate valve on each air riser. Steam used varies from 1 to 5% of total condensation. All radiator connections as shown.

Air Line Systems with Gravity Circulation. The *Paul Air Line System* can be applied to either one or two pipe radiators, and provides a positive and constant means of removing the air from the radiators, without the escape of steam. See Figs. 14, 15 and 16 and Notes under

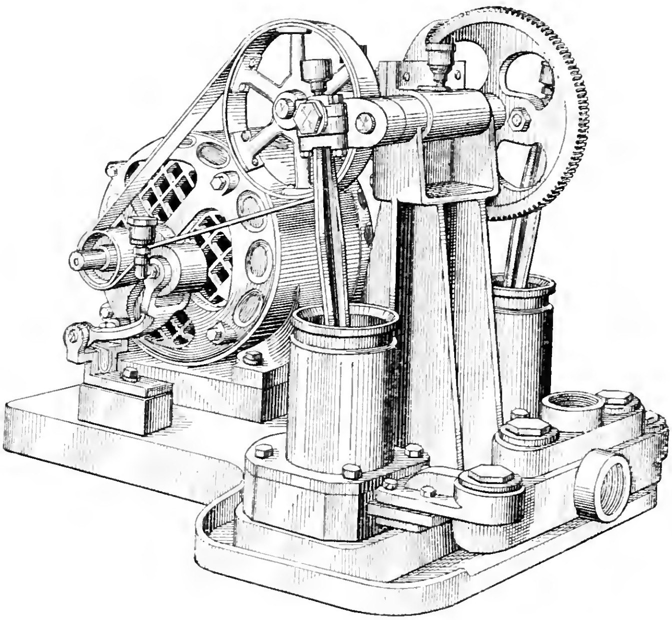


FIG. 17. BELTED COMPRESSOR.

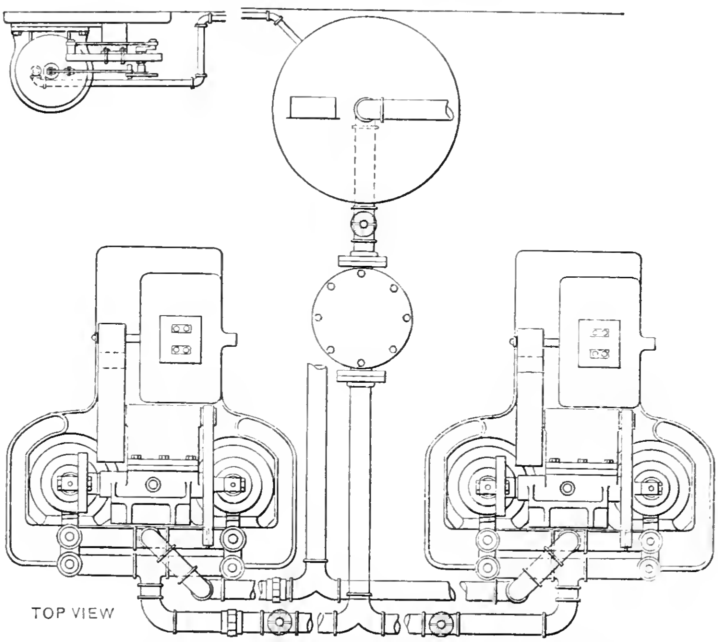


FIG. 18. ELECTRIC VACUUM PUMPS, AND CONNECTIONS.

The above illustration shows method of cross-connecting two "Reliable" Electric Vacuum Pumps for use with heating systems having a larger amount of radiation than one pump can handle. Duplex installations are also used where a reserve unit is desired.

same. Either a steam jet or positive pump type of exhauster must be used for exhausting the air piping and radiators of air so that the steam, at practically atmospheric pressure, will readily fill the entire radiator before closing the automatic air valve, which prevents its escape into the air piping. All odors due to foul air escaping into the rooms are done away with in this system.

It will be seen that the exhauster used on the air removal system is the most essential feature of this system and must be adapted to the means most readily available for its operation.

The use of a positive pump type of exhauster must be resorted to whenever high pressure steam is not available to operate a jet exhauster. Either a direct acting piston pump, using water for operating the driving piston (Fig. 20), or a belted power pump (Fig. 17) may be used.

TABLE 1

SPECIFICATIONS AND DIMENSIONS OF "RELIABLE" ELECTRIC AIR LINE VACUUM PUMPS

No. of Pump	Max. Cap. Sq. Ft. Direct Radia.	Cylinder Sizes		Size of Connection		Strokes per Minute	Horse Power	Dimensions			Shipping Weight Including Equipment
		Bore	Stroke	Disch'e Pipe	Suction Pipe			Height	Width	Depth	
		Ins.	Ins.	Ins.	Ins.			Ins.	Ins.	Ins.	Lbs.
1279	4,000	2¼	3	1	1	150	⅙	28¾	15¼	32	335
112	10,000	3	3½	1	1	70	½	22	25	31½	551
113	18,000	4	3½	1¼	1¼	70	¾	24¼	25	31½	600
114	28,000	4	5	1½	1½	68	1	31	31½	31½	790
115	35,000	5	5	2	2	60	2	31	31½	31½	998

The following figures, and dimension sheets illustrate both water driven and power driven exhausters made by the *Bishop-Babcock-Becker Co.*, and the method of installing them in a complete system for low-pressure steam heating.

A belted compressor is generally installed in the larger systems and a reserve unit (Fig. 18) is often provided to insure continuous service.

TABLE 2

DIMENSIONS AND CAPACITIES OF "RELIABLE" HYDRAULIC VACUUM PUMPS

Number of Pump	Diam. Motor Cylinder Inches	Diam. Suc. Cyl. Inches	Length Stroke Inches	Height Over All Inches	Shipping Weight Including Equipment	Number of Pump	City Water Press'e Pounds	Max. Cap. Square Feet Direct Rad.	Number Pump	City Water Press'e Pounds	Max. Cap. Square Feet Direct Rad.
101	2	2½	4	25	110 lbs.	101	20	700	104	40	4,000
102	2	2½	4	25	140	101	40	800	106	20	6,600
104	2½	4	6	29	156	102	20	900	106	40	9,600
106	2⅜	5 13/16	10	40	200	102	40	1,100	2-106	20	14,000
...	...	...	..	..	......	104	20	2,500	2-106	40	20,000

When city water pressure is between 15 and 20 pounds use pump one size larger than that called for at pressure of 20 pounds.

The method of installing an electric vacuum pump of the preceding type for an air line system in connection with a low-pressure steam heating plant is shown in Fig. 19.

The water driven type of exhauster for small and medium sized systems is shown in Fig. 20 as well as the method of connecting same to the air and water lines. The table of dimensions and capacities following gives the limiting water pressures under which these pumps will operate.

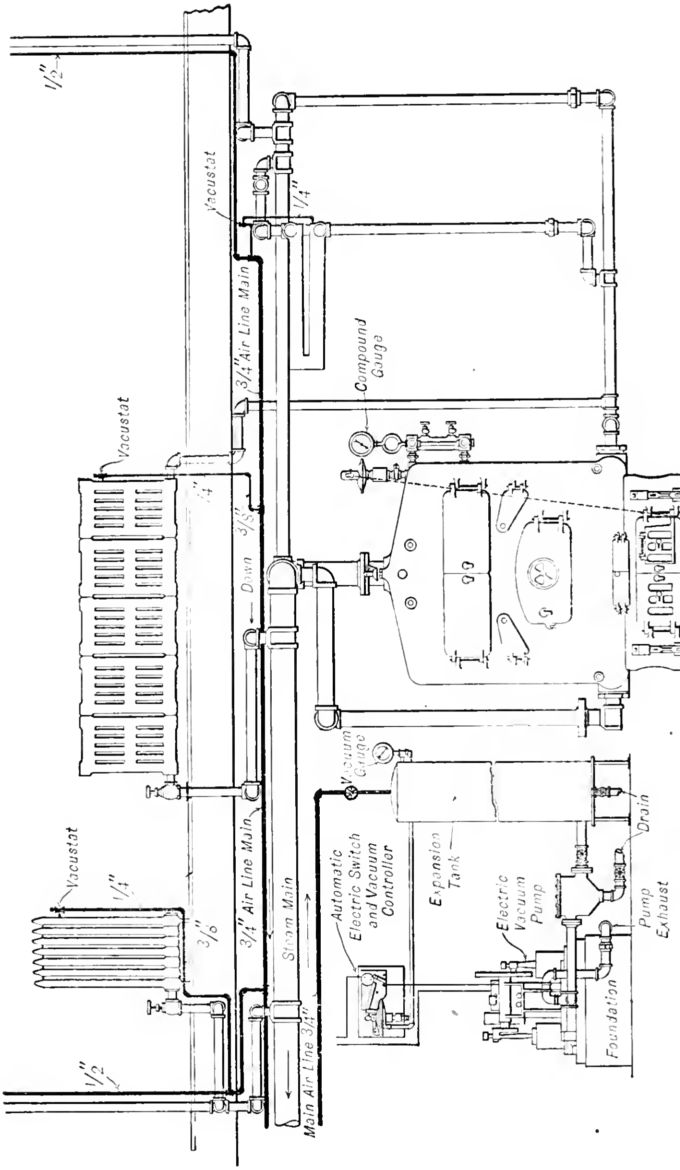


FIG. 19. BISHOP-BABCOCK-BECKER AIR LINE SYSTEM.

Air Line Valves. Automatic air line valves are similar to ordinary automatic air valves except that they do not provide for automatic closing in case water from the radiator enters the valve. These valves may be of the expansion post or expansion chamber type, and two of the latter are shown below.

The *Vacustat* made by the *Bishop-Babcock-Becker Co.* is shown in section in Fig. 21 and its application has been shown in Fig. 19. In operation air or steam enters at the inlet and passes down through the strainer which is just under the removable cap and is of brass, silver plated.

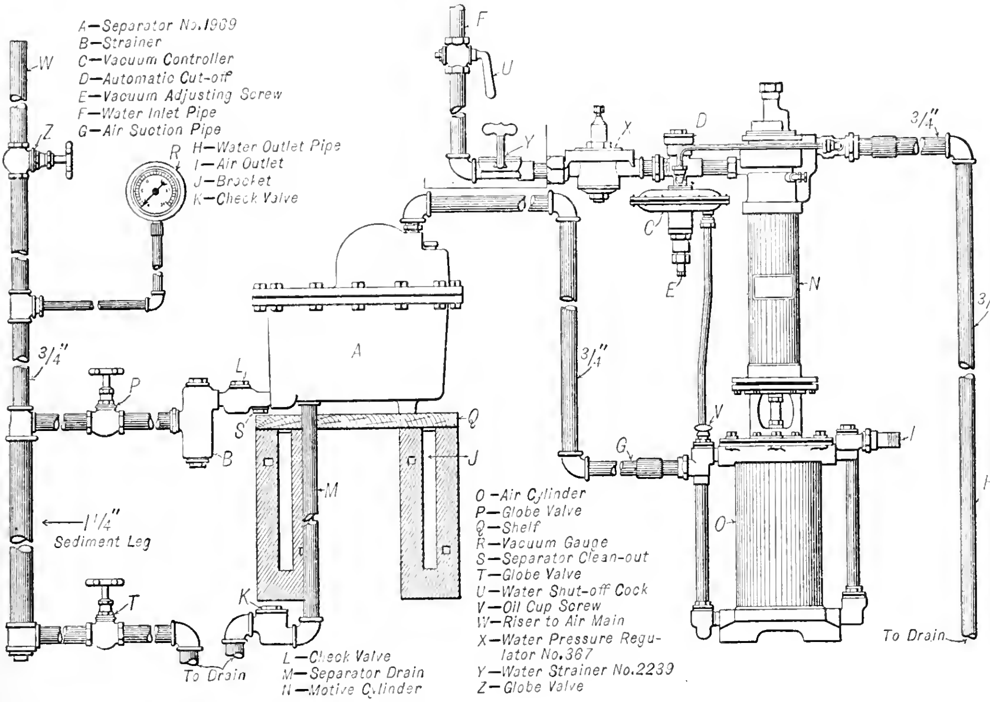


FIG. 20. BISHOP-BABCOCK-BECKER WATER-DRIVEN EXHAUSTER AND CONNECTIONS.

Below this strainer are the air or steam ports which discharge directly against the expansion chamber made of flexible brass and phosphor bronze, and filled with a volatile liquid. When air is passing through the valve this chamber, or thermal member, contracts and uncovers the flexible composition seat just below it so that the air may pass down through the outlet to the air line. When steam reaches the thermal member it expands and closes the passage so that no steam may escape from the radiator. The capacity of the Vacustat is given as 300 square feet of direct radiation.

The *Dunham Air Line Valve* (Figs. 22 and 23) is much smaller but similar in construction to the *Dunham* radiator trap, and is constructed of phosphor bronze. The thermostatic chamber is of corrugated metal and carries a flat disc with a ⅛" lift. The valve port is $^5/_{32}$" in diameter, and the pipe connection to radiator is ⅛", and ¼" to air line. The pressures must not exceed ten pounds. This valve is applied to the radiator in the same manner as the Vacustat as shown in Fig. 19.

Vacuum Systems with Gravity Circulation. The *Trane Mercury Seal Vacuum System* is also an air line system, and can be used on one or two pipe radiators as shown for the Paul System in Figs. 14 and 16. The essential feature is a mercury seal (Fig. 24, and Notes) which is attached to the end of the air main, and readily permits the escape of air from the system when a few pounds pressure is generated in the steam mains, but which interposes a mercury column 24″ high if any air attempts to re-enter the system. Hence by raising pressure enough to drive out the air occasionally this system may be operated as a vacuum system at a *variable temperature* corresponding more nearly to the heating requirements. Air valves are used at the radiators for connecting the air line to prevent the escape of steam into the latter. It must be noted that unless all radiator and air valves, fittings, etc., are absolutely tight it will be impossible to hold any vacuum at all, and the system will fail to operate below atmospheric pressure.

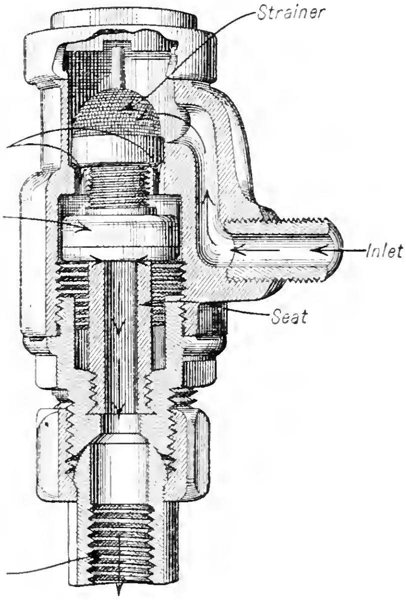
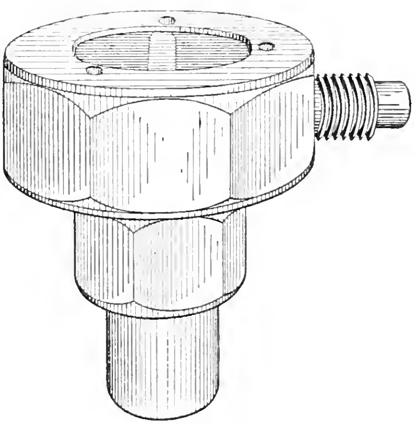


Fig. 21. Section of Vacustat.

Vapor Systems with Gravity Circulation. The *Moline Vapor System*, as the name indicates, operates with *vapor* or steam at atmospheric pressure. (Fig. 25 and Notes.) This is a two-pipe system with dry return, in which a very low pressure steam jet, supplied with steam from the end of the steam main, and exhausting into a condenser coil equipped in turn with an air trap and vacuum valve is used to free the return system of air, but permits the condensation to flow back to the boiler by gravity. The radiators are of the water type and equipped with a graduated supply and special return valve, having an adjustable sleeve with ports so proportioned as to prevent the escape of steam into the return piping.

The *Webster Modulation System*, which is similar to the Moline System, is shown in Fig. 26.



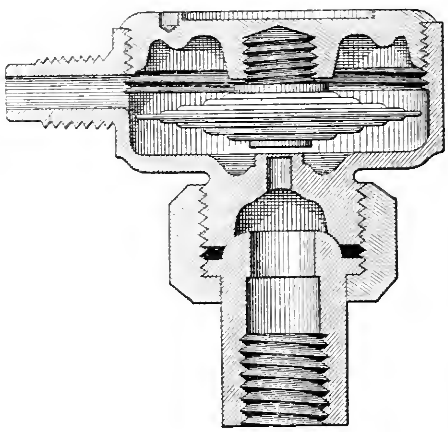


Fig. 22. Dunham Air Line Valve. Fig. 23.

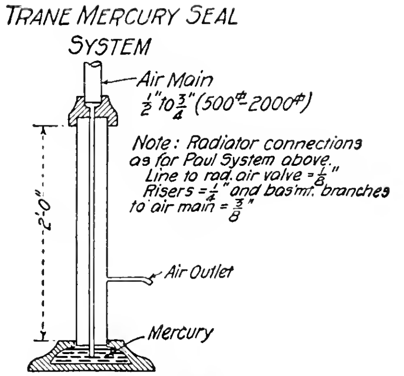


FIG. 24.

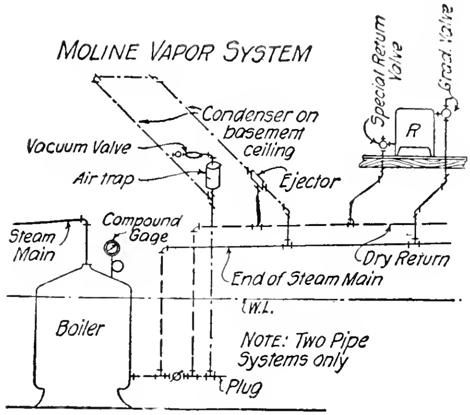


FIG. 25.

NOTES.

TRANE SYSTEM: Air in this system driven out periodically by steam pressure through the mercury seal, which will then maintain a vacuum as high as 24" of mercury. Seldom used on systems above 2,000 sq. ft. Steam and return piping as for gravity work.

MOLINE SYSTEM: Essentially a two-pipe low pressure steam system, two valves to each radiator. Live steam from supply mains pulls air from return main through an ejector, no seals on return risers except special valve at radiator.

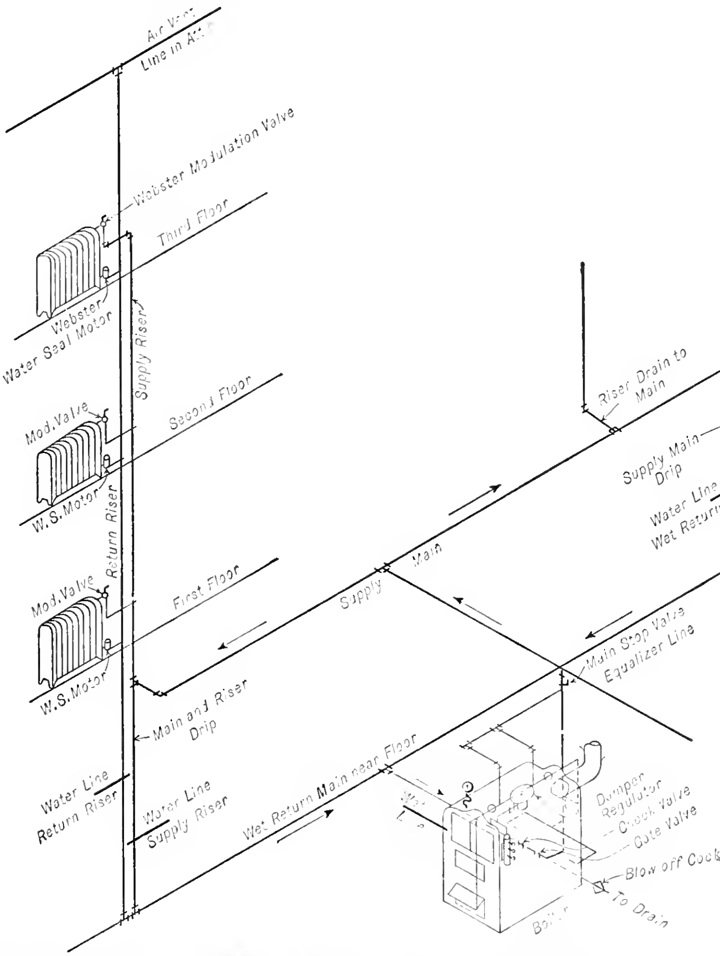


FIG. 26 TYPICAL PIPING ARRANGEMENT WEBSTER MODULATION SYSTEM WITH SEALED WET RETURN.

MECHANICAL STEAM HEATING SYSTEMS

The low pressure mechanical circulating systems, as already explained, depend on *pumps* or *traps* for the return of water of condensation to the boiler. These pumps may be of the steam or motor driven type and are usually under the automatic control of a pump governor, which maintains a constant water line in the receiving tank. The arrangement of such a system is shown in the chapter on "Exhaust Steam Heating," where it is assumed that high pressure steam is available for operating the steam pump. Pump and receiver sizes are given in Table 3, or if desired the size of receiver may be based on the assumption that its capacity must be sufficient to hold the steam condensed by the entire system in 20 or 30 minutes. Tables of pump sizes will also be found in the chapter on "Power Plants."

TABLE 3

DIMENSIONS AND CAPACITIES OF *BURNHAM* AUTOMATIC CONDENSATION PUMP AND RECEIVER

Number	Size of Pump, Inches	Size of Receiver, Inches	Gals. Del'd per Min.	Sq. Ft. Radiating Surface Pump Will Drain	Space Occupied, Inches			Shipp'g Weight, Pounds (Approximate)
					Lgth	Width	H'ght	
0*B*	3 x $1\frac{3}{4}$ x 3	Oval	4	2,000	20	20	30	350
1*B*	3 x 2 x 3	Oval	5	2,500	20	20	30	350
2*B*	4 x $2\frac{1}{2}$ x 5	Oval	10	5,000	26	24	33	575
4*B*	5 x 3 x 6	15 x 26	15	7,500	41	28	38	900
5*B*	$5\frac{1}{2}$ x $3\frac{1}{2}$ x 7	19 x 28	25	12,500	41	28	38	1,250
7*B*	$6\frac{1}{8}$ x 4 x 8	19 x 28	35	17,500	49	30	38	1,360
8*B*	$7\frac{5}{8}$ x $4\frac{1}{2}$ x 9	24 x 38	50	25,000	49	32	40	2,150
9*B*	$8\frac{1}{2}$ x 5 x 10	24 x 38	65	32,500	56	32	42	2,350
10*B*	10 x 6 x 12	21 x 50	100	50,000	68	38	42	3,000
11*B*	12 x 8 x 12	21 x 50	155	77,500	71	38	42	3,500

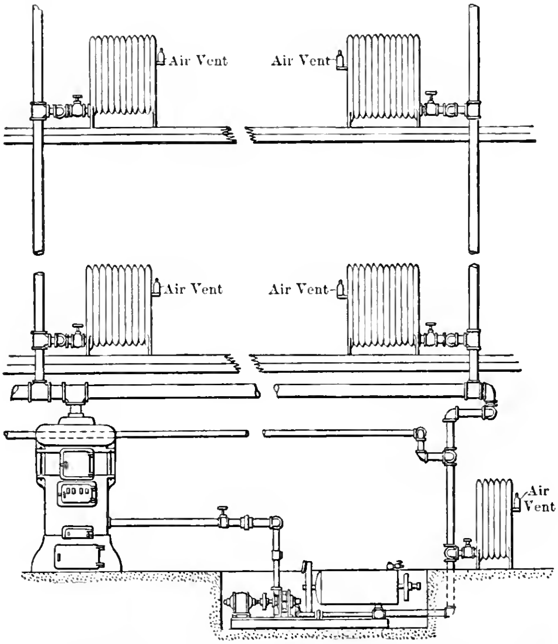


FIG. 27. AUTOMATIC CONDENSATION PUMP FOR ONE-PIPE SYSTEM.

If high pressure steam is not available then electrically operated power or centrifugal pumps may be used and also placed under automatic control.

Applications of an automatic condensation pump and receiver to practice are shown in Figs. 27 and 28, which show, first a single pipe system with the radiators below the water line of boiler, and second a two pipe system with graduated supply and automatic vacuum return valves on radiators.

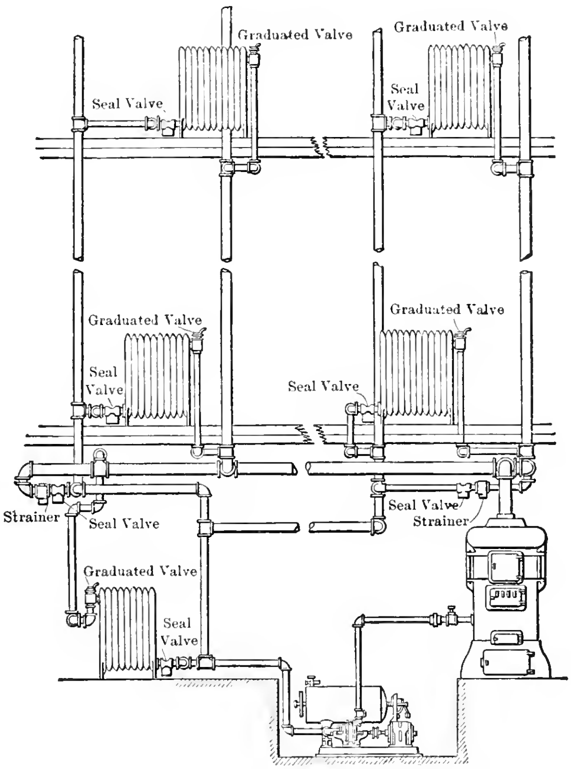


Fig. 28. Automatic Condensation Pump—Two-Pipe System.

In each of the above cases it will be noted that by placing the automatic pump and receiver in a small pit it has been possible to place the boiler on the basement floor and avoid the use of a large boiler pit, which would be necessary if a gravity circulating system had been installed.

It must be remembered, however, that the cost of the pump, receiver and motor, as well as the cost of operation of the latter, must be taken into account before any "saving" in cost of boiler pit is allowed for. The gravity circulating system with boiler pit is almost invariably much cheaper to install and operate than any automatic pump and receiver system where electric current must be purchased.

Condensation Pumps for Mechanical Systems. Automatic *electric driven condensation pumps*, equipped with tilting receivers for opening and closing the controlling switch on the motor circuit, are used on systems where radiators are on the same or lower level than the boilers, or where the pressure in the boiler is higher than in the heating mains, by the amount given in Table 4.

Dimension drawings (Figs. 29 and 30) together with Table 5 give the general dimensions and details of this equipment as manufactured by the *Chicago Pump Co.*, Chicago, Illinois.

The kind of current available must be specified and voltage stated in order that the proper motor may be supplied.

TABLE 4

CHICAGO CONDENSATION PUMPS (STEEL BASE)

Number	Maximum Sq. Ft. Direct Radiation	H.p. Motor	Approx. Shipping Weight	*Boiler Pressure
1	1,500	$\frac{1}{4}$	300	7 lb.
3	6,000	$\frac{1}{2}$	600	10
4	10,000	$\frac{3}{4}$	700	15
5	15,000	1	800	20
6	25,000	2	900	20

* These pumps will operate against a boiler pressure equal to the pressure for which they are listed plus the pressure on the return pipe at receiver.

Only condensation pumps of 15,000 sq. ft. capacity and over include the auxiliary tank. (Fig. 29.)

These pumps are fitted with outboard ring-oiled bearings. Other condensation pumps depend upon the water they handle to lubricate their bearings, but water, especially hot water, is a poor lubricant; it will cause the bearings to wear rapidly and often to stick, which may cause damage to motor and possibly result in its burning out.

This pump also is fitted with receiving and settling chamber with plugged openings, for cleaning; and all parts of the pump are mounted on one iron base, connected up, adjusted and ready for piping and wiring connections.

The automatic controlling switch is of the double carbon contact type, and is controlled by a heavy galvanized iron tilting receiving tank and not by a float. Floats are liable to crack and become water-logged.

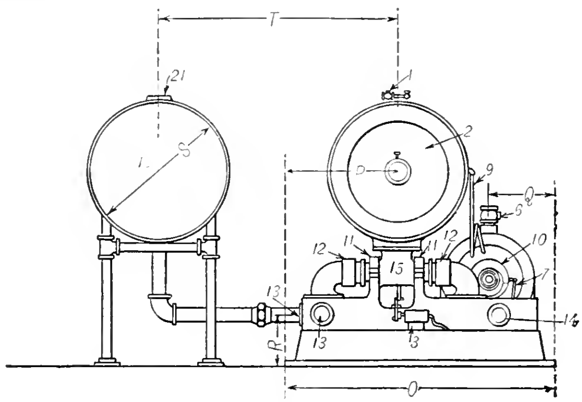


FIG. 29.

PUMP PARTS

1. Air Cock and Relief Valve.
2. Counter Weight.
3. Automatic Boiler Feed Valve.
4. Flexible Coupling.
5. Connection to Automatic Switch.
6. Discharge of Pump.
7. Boiler Feed Regulating Valve.
8. Packing Glands.
9. Steam Tube.
10. Outboard Ring-oiled Bearing.
11. Babbitt Bearings for Tilting Tank.
12. Trunnions and Packing Glands.
13. Inlets and Clean-outs.
14. Clean-out only.
15. Brass Tee for Tilting Tank.
16. Auxiliary Tank.
17. Auxiliary Tank Inlet.

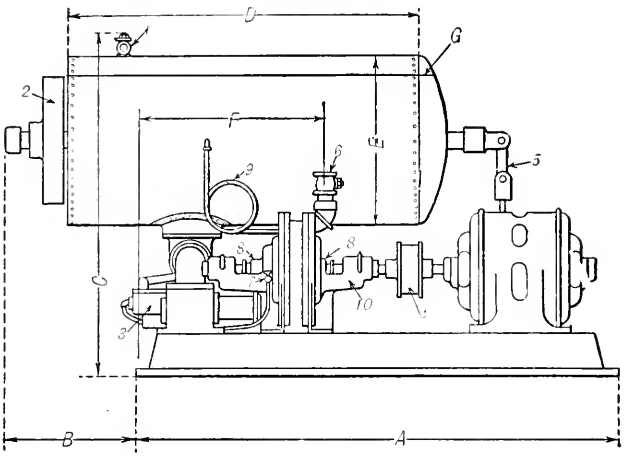


FIG. 30.

TABLE 5

DIMENSIONS OF *CHICAGO* CONDENSATION PUMPS. (See Fig. 30.)

Dimension	25,000 Sq. Ft.*	20,000 Sq. Ft.	15,000 Sq. Ft.	10,000 Sq. Ft.	6,000 Sq. Ft.	3,000 Sq. Ft.	1,000 Sq. Ft.
A	52	52	52	46	46	46	30
B	18½	18½	18½	18½	12	9	8
C	38	38	38	33	33	33	26
D	48	48	48	48	36	30	30
E	16	16	16	16	16	16	12
F	19	19	19	19	17 11/16	17 11/16	4
G	30	30	30	30	30	30	24
O	32	32	32	28	28	28	21
P	10⅝	10⅝	10⅝	10⅝	10⅝	10⅝	9
Q	9	9	9	9	9	9	3
R	9	9	9	7½	7½	7½	4¾
S	24	18	16	....	....	....	....
T	30	27	26	....	....	....	....
Length of Auxiliary Tank	60	60	48	....	....	....	....

*NOTE.—This represents sq. ft. of Direct Radiation condensing 0.25 lb. steam per sq. ft. per hour.

Tilting Traps. Tilting traps are often used for discharging and returning condensation to receivers or boilers. There are two principal types of tilting traps in which the filling and emptying of a receiving tank cause it to operate the valves of the trap.

One of these is known as a *non-return* trap, and is constantly subject to the pressure from the lines being drained, and the other is the *return* trap, which periodically admits steam at full boiler pressure above the water in the trap when the trap tank tilts.

TABLE 6

OVER-ALL DIMENSIONS OF NON-RETURN TILTING TRAPS

(*Morehead Co.*)

Number	Height		Width		Length	
	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.
1	2	¼	1	4½	2	7¾
2	2	4½	1	5½	2	11
3	2	10	1	9	3	8
4	3	½	1	9½	4	3½
5	3	3½	2	6	4	10½
6	....		....		....	

In operation the *non-return trap* fills while in a horizontal position as shown in Fig. 31, and when the tank is nearly full the counterweight is overbalanced and the trap tilts into the position shown in Fig. 32, opening the discharge valve at the same time. The pressure in the line being drained must be sufficient to force the water out until only enough condensation remains to seal

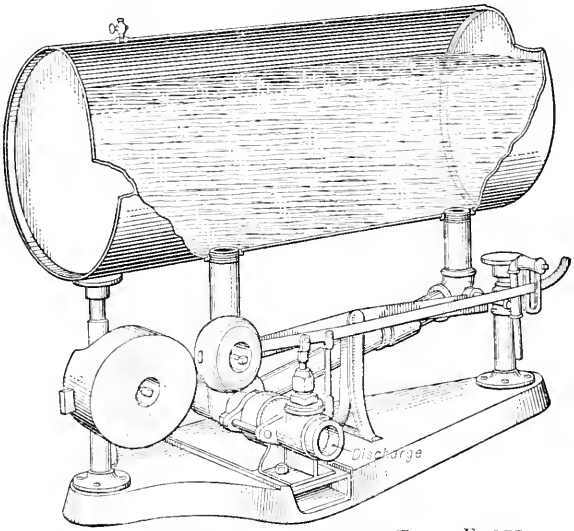


FIG. 31. MOREHEAD NON-RETURN TRAP—FILLED.

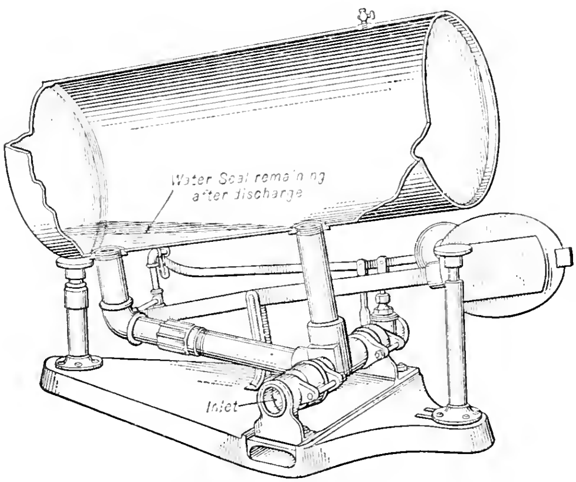


FIG. 32. MOREHEAD NON-RETURN TRAP—EMPTY

the discharge outlet, when the trap tilts back and closes the discharge valve preparatory to refilling. By reference to Fig. 33 it will be seen that the trap fills through the forward opening in the tank and discharges at the rear, when the lever-actuated globe valve is opened. A swing check should be installed in the discharge line beyond the trap.

The operation of the *return trap* is best understood by reference to Fig. 34, from which it will

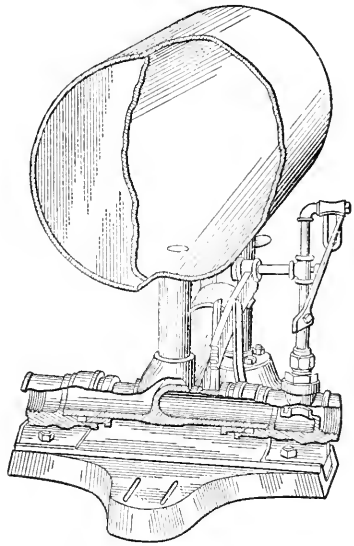


FIG. 33. MOREHEAD TILTING NON-RETURN TRAP.

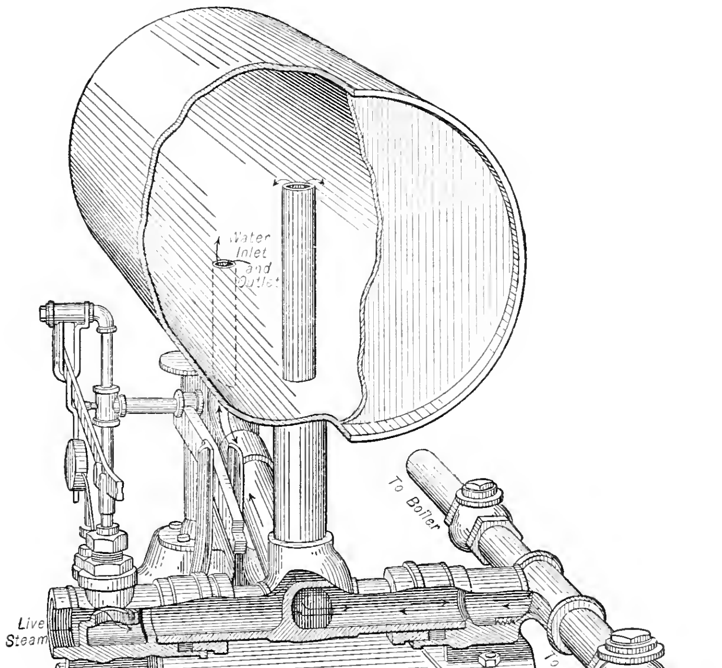


FIG. 34. MOREHEAD RETURN TRAP.

be seen that water enters and leaves the trap through the right-hand trunnion under the control of two check valves, one of which opens toward the trap to permit the condensation to enter the tank, and the other toward the boiler to permit the discharge of water whenever the pressure in the tank is great enough to equalize the boiler pressure and also close the first check valve. Steam enters through the globe valve just beyond the left-hand trunnion, and is delivered at a point above the water in the tank whenever the tank tilts downward, due to filling.

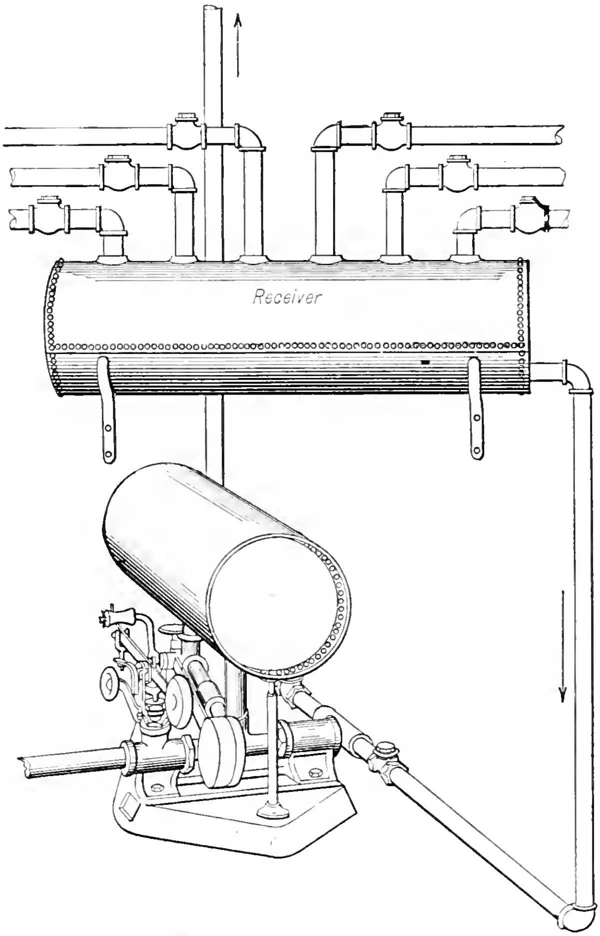


FIG. 35. INSTALLATION OF TILTING RETURN TRAP.

Hence, if this trap is placed four feet above the boiler water line, and the steam connection is made to this same boiler, the tilting of the trap will cause the pressures to be equalized above and below the water in the tank, and the condensation will flow into the boiler under a four-foot head. A second trap may be required to get the water into the tank above the boiler. The application of this tilting, or return trap, to a typical installation is shown in Fig. 35.

The tilting condenser trap (Fig. 36) is simply a modification of the return trap in which a water spray is connected into the tilting tank chamber to serve as a jet condenser. This spray

is automatically supplied with water, just as the tank tilts back to its filling position, so that immediate condensation of the steam in the tank is brought about, and a partial vacuum formed causing the tank to "pull" strongly on the return lines. Just as soon as the tank is filled and

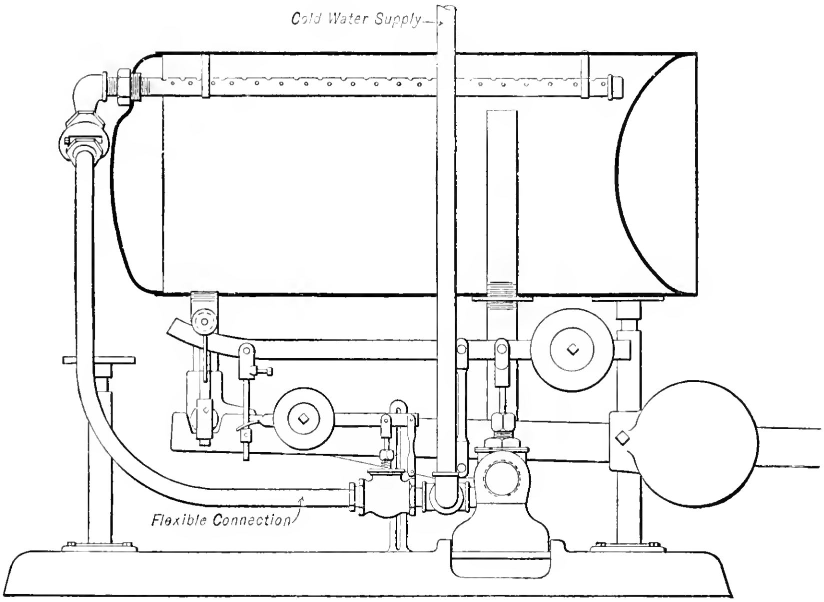


FIG. 36. MOREHEAD TILTING CONDENSER TRAP.

tilts to admit live steam, this water is shut off and the discharge takes place under boiler pressure.

All tanks are equipped with air valves. Capacities, pipe connections and weights of these traps are given in the tables following:

TABLE 7

MOREHEAD TILTING NON-RETURN STEAM TRAP

Number	Inlet, Inches	Outlet, Inches	Capacity in Water Discharged per Hour	Drainage Capacity in 1 Inch Pipe Lineal	Capacity Square Feet Direct Radiation	Capacity Lineal Feet Hot Blast Heater	Weight
			Gal.	Ft.			
21............	1	1	200	12,000	3,000	1,300	100
22............	1¼	1¼	400	25,000	5,200	2,400	175
23............	1½	1½	600	40,000	12,000	5,200	250
24............	2	2	720	60,000	21,000	9,000	275
25............	2½	2½	900	90,000	33,000	16,000	350
26............	3	3	1300	140,000	50,000	50,000	450

TABLE 8

MOREHEAD TILTING RETURN, VACUUM AND CONDENSER STEAM TRAPS

Number	Size of Drum	Size of Inlet and Outlet Connections, Inches	Size of Steam Pipe Connections, Inches	Capacity of Water in Lb. per Hour	Drainage Capacity in Feet of 1 Inch Pipe Lineal	Capacity Square Feet Direct Radiation	Capacity Lineal Feet Hot Blast Heater	Weight
1	10 x 24	1	1	1,050	5,000	2,300	1,000	100
2	12 x 30	1 1/4	1	1,850	9,000	4,000	1,800	175
3	14 x 36	1 1/2	1 1/4	4,000	20,000	9,000	4,000	250
4	16 x 40	2	1 1/4	6,000	35,000	16,000	7,000	275
5	18 x 42	2 1/2	2	11,000	50,000	25,000	12,000	350
6	18 x 42	3	2	15,000	75,000	40,000	18,000	400

NOTE.—The above capacities are figured on a basis of 50 pounds pressure to the square inch. The above drainage capacity in inch pipe is based on ordinary radiating conditions. For lumber kilns, greenhouses and moist goods, divide by two. For laundries, brick dryers and wet goods, divide by three. For fan stacks and blowers, divide by five.

3 feet of 1-inch pipe equals one square foot of surface. 2.3 feet of 1 1/4-inch pipe equals one square foot of surface. 2 feet of 1 1/2-inch pipe equals one square foot of surface. 1.61 feet of 2-inch pipe equals one square foot of surface.

TABLE 9

MOREHEAD RECEIVERS

Number	Length, Inches	Height, Inches	Diameter, Inches
1	30	16	10
2	40	20	12

NOTE.—No. 1 Receiver has capacity for Traps Nos. 1 and 2. No. 2 Receiver has capacity for Traps Nos. 3, 4, 5 and 6.

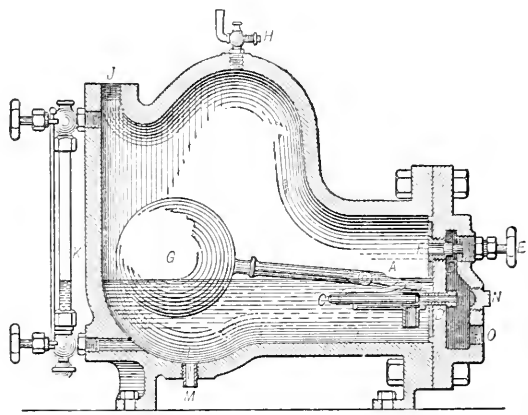


FIG. 37. AMERICAN IDEAL, NON-RETURN TRAP FOR LOW PRESSURES.

A. Powerful valve mechanism.
C. Valve stem.
D. Valve seat.
EE. By-pass valve and outlet.
F. By-pass valve seat.
G. Float.
H. Air cock.
J. Inlet.
K. Water gage.
L. Sediment chamber.
M. Blow-off for sediment.
N. Plug to valve seat.
O. Outlet.

Float Traps. A non-return float trap is often used for discharging condensation to a vented receiver, open feed-water heater, or the sewer. A sectional view of such a trap made by the

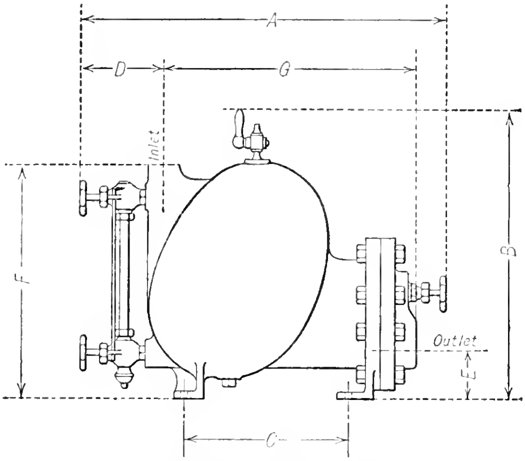


FIG. 38. AMERICAN IDEAL STEAM TRAP.

American Steam Gage and Valve Mfg. Co. is shown in Fig. 37 and capacities at varying low pressures from 1 to 30 lb. for all sizes are listed in Table 11. The floats used are heavy, designed to operate on pressures from 1 to 125 pounds, and exert a powerful leverage to open or close the discharge valve, which is the full diameter as given in the table.

TABLE 10

DIMENSIONS OF AMERICAN IDEAL STEAM TRAP (See Fig. 38)

Size	A	B	C	D	E	F	G	Width Required	Pipe Conn.
No.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
1..........	22¾	17¼	11	5	3⅝	15¼	16	10	½
2..........	24	19¼	11¼	5	3⅝	16½	16½	10¾	¾
3..........	27½	21	13	5½	4 1/16	18	19⅜	12⅛	1
4..........	30	22	15	5¾	4	19⅞	21⅝	12¾	1¼
5..........	33½	25	17½	6¼	3¾	23⅞	23⅜	14	1½

This trap is also designed to operate on pressures up to 250 pounds per sq. in., giving much greater capacities than listed in Table 11. See maker's catalog.

Mechanical Vacuum Systems. The so-called mechanical "*Vacuum Systems*" are practically all of the two-pipe type, and have a vacuum pump attached directly to the returns. See systems shown in chapter on "Exhaust Steam Heating." This pump may be steam or motor driven, but must be capable of handling both air and water at fairly high temperatures.

TABLE 11

RATED CAPACITIES PER HOUR OF AMERICAN IDEAL STEAM TRAP—MODEL "C"

Low Pressure (1 30 Lbs.) *American Steam Gage & Valve Mfg. Co.*

May 5, 1914

Size of Trap		Pounds per Square Inch Gage																		
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	20	25	30	
No. 1 Trap ½" Opening	Gallons of water . . .	235	335	405	475	525	580	630	670	715	755	790	825	860	890	925	1065	1185	1300	No. 1 Trap Pipe Connections ½" I. P. S.
	Pounds of water . . .	1960	2800	3380	3970	4380	4850	5265	5600	5975	6320	6600	6900	7180	7430	7730	8880	9900	10880	
	Lineal ft. of 1" Pipe	3480	4970	6000	7050	7780	8620	9350	9950	10600	11220	11700	12350	12750	13200	13720	15780	17600	19300	
	Sq. feet of Rad	1160	1660	2000	2350	2590	2873	3170	3320	3520	3740	3890	4070	4235	4380	4570	5260	5870	6430	
No. 2 Trap ¾" Opening	Gallons of water . . .	350	500	610	710	795	870	940	1004	1067	1120	1184	1233	1287	1332	1380	1580	1770	1950	No. 2 Trap Pipe Connections ¾" I. P. S.
	Pounds of water . . .	2920	4180	5100	5935	6640	7265	7850	8380	8910	9360	9880	10315	10750	11140	11550	13200	14800	16300	
	Lineal ft. of 1" Pipe	5180	7430	9050	10550	11600	12900	13950	14900	15840	16650	17550	18330	19100	19800	20500	23470	26300	29000	
	Square feet of Rad .	1720	2470	3010	3500	3870	4280	4635	4970	5282	5550	5850	6115	6375	6600	6830	7820	8770	9660	
No. 3 Trap 1" Opening	Gallons of water . . .	570	810	990	1150	1260	1410	1522	1623	1725	1810	1908	1990	2074	2150	2230	2580	2880	3150	No. 3 Trap Pipe Connections 1" I. P. S.
	Pounds of water . . .	4730	6770	8270	9620	10530	11780	12725	13550	14100	15120	15930	16620	17320	17980	18600	21550	24100	26300	
	Lineal ft. of 1" Pipe	8400	12040	14700	17080	18740	20900	22600	24300	25550	26900	28300	29500	30750	31920	33090	38300	42800	46750	
	Square feet of Rad .	2800	4020	4900	5690	6250	6970	7530	8100	8517	8970	9430	9830	10250	10650	11000	12770	14260	15580	
No. 4 Trap 1¼" Opening	Gallons of water .	870	1270	1520	1765	1940	2170	2340	2490	2650	2800	2930	3060	3190	3310	3410	3970	4420	4850	No. 4 Trap Pipe Connections 1¼" I. P. S.
	Pounds of water . . .	7270	10600	12700	14750	16200	18150	19550	20800	22140	23100	24500	25600	26650	27700	28700	33150	36900	40500	
	Lineal ft. of 1" Pipe	12930	18860	22600	26200	28800	32200	34700	36900	39300	41650	43500	45500	47300	49250	51000	59000	65650	72000	
	Square feet of Rad .	4310	6260	7530	8730	9600	10730	11570	12300	13100	13880	14500	15150	15760	16400	17000	19660	21880	24000	
No. 5 Trap 1½" Opening	Gallons of water . . .	1240	1768	2140	2500	2800	3070	3320	3530	3750	3970	4150	4330	4525	4685	4870	5620	6270	6850	No. 5 Trap Pipe Connections 1½" I. P. S.
	Pounds of water . . .	10360	14770	17750	20900	23400	25870	27760	29500	31350	33150	34700	36200	37800	39120	40700	47000	52300	57250	
	Lineal ft. of 1" Pipe	18320	26270	31550	37130	41650	46000	49250	52400	55650	59000	61650	64250	67150	69500	72700	83500	93000	102000	
	Square feet of Rad .	6100	8750	10510	12365	13880	15340	16400	17450	18535	19660	20530	21400	22380	23170	24230	27830	31000	34000	

The selection of a *vacuum pump* is usually based on the sq. ft. of radiation to be drained or pounds of condensed steam to be handled per hour, and the principal dimensions are given in the following table:

TABLE 12

CAPACITY AND SIZE OF STEAM-DRIVEN VACUUM PUMPS

Steam Cylinder Diam., Inches	Water Cylinder Diam., Inches	Stroke Inches	Steam Pipe	Exhaust Pipe	Suction Pipe	Discharge Pipe	Draining Capacity Sq. Feet, Direct Radiation*	Draining Capy. Lbs. Condensed Steam Pr. Hr.	Floor Space, Inches
3	2 5/8	4	3/8	1/2	1 1/4	1 1/4	2700	810	30 x 6
4	4	6	1/2	3/4	2	1 1/2	7000	2100	40 x 10
6	6	10	1	1 1/4	4	3	16000	4800	59 x 14
10	10	12	1 1/4	2	6	5	40800	12240	72 x 20
10	12	12	1 1/4	2	8	7	62000	18600	72 x 20
10	14	12	1 1/4	2	10	8	85000	25500	......
10	16	18	1 1/2	2	12	10	92000	27600	......
10	18	18	1 1/4	2	12	10	128000	38400	......

NOTE.—Vacuum pumps with belted electric motors (Fig. 17) are made by the *Bishop-Babcock-Becker Co.* with capacities of 2,000, 5,000, 10,000, 17,000 and 25,000 sq. ft. of direct radiation.
* Condensation figured at 0.3 lb. per square foot radiating surface per hour.

This pump should be under the control of a reliable *vacuum pump governor* so that when the required vacuum has been produced the pump will stop.

Radiator Traps for Vacuum Systems.* The distinguishing characteristic of these mechanical "Vacuum Systems" is found in the *trap* on the return end of the radiator, and this trap almost invariably gives the system its name, such as *Dunham, Webster, Illinois,* etc. As a matter of fact, these systems are all alike, although the various traps now on the market differ somewhat in construction, in principle of operation, and in efficiency.

A radiator trap for a vacuum system is supposed: (1) to prevent the escape of any steam into the return riser, which is not sealed beyond the trap but connected directly into the main return; (2) to allow all the water of condensation and air to escape from the radiator; (3) to be quiet in operation, and (4) to be free from fouling due to scale, core sand, etc., which may tend to prevent proper seating when the trap closes.

In case the trap *fails* in the above respects, we have the following undesirable results:

(1) Steam may escape into the returns, which is a loss, and if sufficient in amount may make it impossible to maintain a vacuum unless a large amount of cold water is injected into the returns just before they are taken by the vacuum pump. (2) Air and water may be held back, should the trap fail to open promptly, and some of the radiating surface rendered inefficient. (3) The mechanism of the trap may be noisy in operation, and in hospital, or general building work this would not be tolerated. (4) Passageways and valve seats must be so designed as to avoid the lodgment of particles of scale or sand on the seat, and thereby hold trap open and possibly score the seat at the same time.

These traps are classified as: (1) Thermostatic traps, with a *volatile liquid* operating an expansive chamber, which in turn opens or closes the valve, or with an *expansion post* of carbon or hard rubber composition, which expands and closes the valve when steam strikes the post: (2) Float traps, in which the collection of water in the body of the trap raises *a float,* thereby allowing liquid to escape while preventing the escape of steam. In these latter types it is almost impossible to arrange for the escape of air and not permit more or less steam to pass out with it.

* NOTE.—Great care must be exercised in selecting the proper sized trap or traps for hot-blast heating coils. The capacity ratings for all traps are in terms of direct cast-iron radiation, on a condensation basis of approximately 0.25 lb. per sq. ft. per hour. Every unit of blast coil must be reduced to that basis before trap sizes are chosen and specified. See the chapter on "Hot Blast Heating" for further details in reference to rating of vacuum traps for hot-blast coils.

The construction sizes, capacities, and method of operation of typical vacuum traps are shown in Figs. 39 to 52.

Illinois Radiator Return Traps. The *Illinois Engineering Co.* makes a *thermostatic vacuum trap* or valve (Figs. 39 and 40), in which the thermostatic element or bellows is on the discharge side of the trap, and therefore subject to the conditions existing in the return main. The valve is so arranged that the seat is vertical, and is closed by a ball disc. The vertical seat is obviously

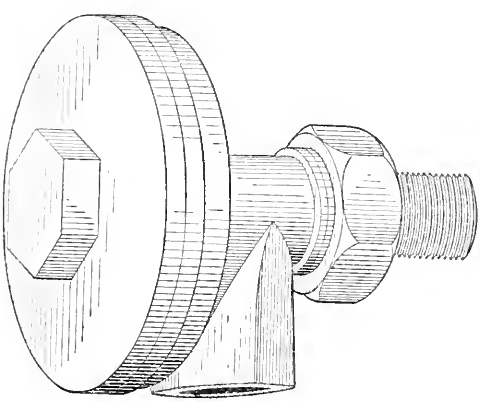


Fig. 39.

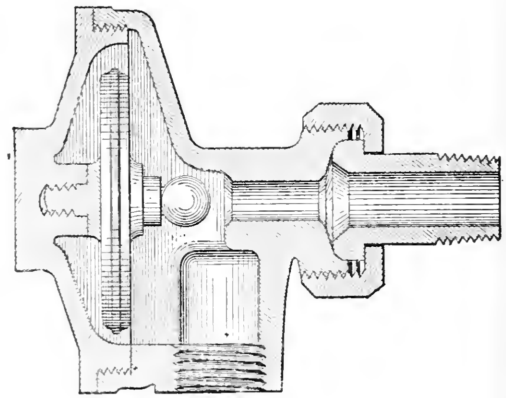


Fig. 40.

Illinois Thermostatic Vacuum Trap.

TABLE 13

CAPACITIES OF ILLINOIS RADIATOR TRAPS

Size In.	Capacity—Square Feet Direct Radiation	Pipe Connections, In.	Weight, Lb.	Diameter of Port, In.
½	150	½	3	5/16
¾	450	¾	4	. .
1	1000	1	17	. .

less liable to fouling than a horizontal seat, and the ball is supposed to adjust itself to a perfect contact with the seat in case of slight misalignment.

This same company also manufactures a *float type* of *vacuum trap* or valve shown in section in Figs. 41 and 42. This trap, as applied to direct radiators (Fig. 41), contains a hollow float *A* with open bottom, which rests at a small angle upon a raised valve seat *B*. It will therefore be evident that when there is no water in the trap air or steam may pass into the return line at all times through a very small opening. If water in considerable amount enters the trap the float is raised, and the entire area of the discharge port becomes available for its escape. A by-pass plug *D* with seat at *E* is provided for blowing out the accumulations retained by strainer *F*.

The *Illinois blast-coil trap* or valve (Fig. 42) operates in a similar manner to the radiator valve just described—except that no by-pass or strainer is provided, and the pipe connections are ¾″ or larger.

Dunham Radiator Return Traps. The *C. A. Dunham Co.* makes a *thermostatic trap* or valve for use on vacuum or vacuo-vapor systems as shown in Figs. 43 and 44, while its method of application is indicated in Fig. 45. In this valve the seat is horizontal, and in the radiator traps

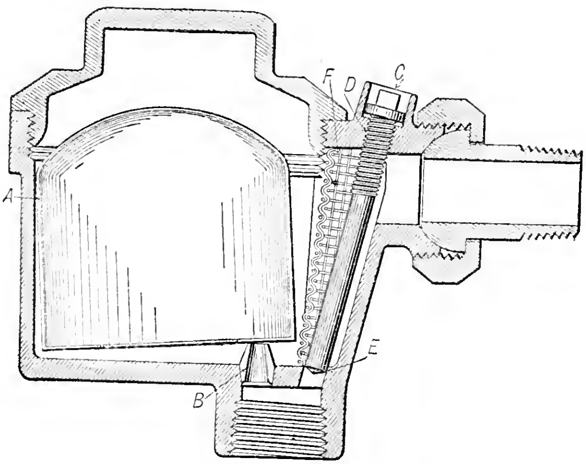


FIG. 41. ILLINOIS FLOAT RADIATOR TRAP.

a flat metal disc, which closes against a raised seat, is secured directly to the expansion bellows (Fig. 44). These smaller sizes are constructed of phosphor-bronze and made in four patterns—right hand, left hand, straight way, and angle.

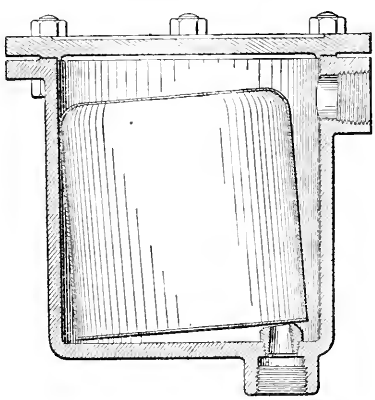


FIG. 42. ILLINOIS FLOAT BLAST TRAP.

TABLE 14

CAPACITIES OF ILLINOIS FLOAT TRAPS

Size	Capacity—Square Feet Direct Radiation	Pipe Connections, Ins.	Weight, Lb.	Diameter of Port, In.
0	60	1/2	2	..
1	120	1/2	3	1/8
2	200	1/2	5	..
3	300	1/2 or 3/4	10	..
4	450	1/2 or 3/4	10	..
6	900	3/4	25	..
8	1700	1	45	..
10	2500	1 1/4	80	..
12	4500	1 1/2	135	..

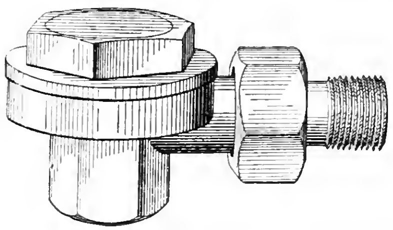


FIG. 43.

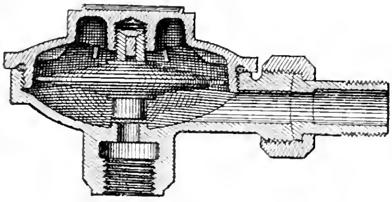


FIG. 44.

DUNHAM THERMOSTATIC TRAPS.

TABLE 15

CAPACITIES OF DUNHAM TRAPS

Size Nos.	Size In.	Capacity, Direct Radiation, Sq. Ft.	Pipe Connection, In.	Weight, Lb.	Diameter of Port, In.	Lift, In.
1	1/2	100	1/2	1 1/2	...	..
2	1/2	350	1/2	2 1/2	3/8	1/8
3		450	3/4	...	...	..
B.T.	3/4	1500	3/4	13	3/4	3/16
B.T.	1	3000	1	21	1	..

NOTE.—These traps are designed for steam pressures not in excess of 10 lb. gage. For main and riser drips, use no smaller trap than the No. 3, and install trap as per details Nos. 3, 4, and 5, shown on pages 226-a and -b.

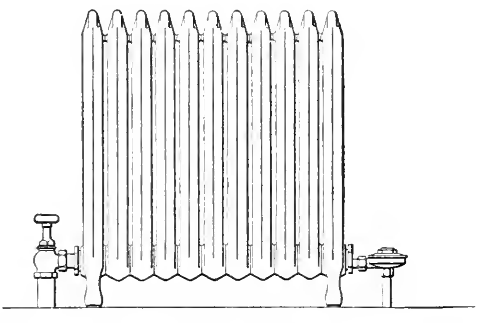


FIG. 45. APPLICATION OF DUNHAM RADIATOR TRAP.

The *Dunham blast trap* (Fig. 46) is equipped with a removable hard rubber disc, which is carried in a self-aligning holder to permit of more perfect closing of the valve should the vertical axis of the bellows not be exactly perpendicular to the valve seat. The body of this trap is made of cast iron in the straight-way and angle types, and a blow-off plug is provided on the inlet side. This trap is of large capacity as indicated in Table 15 and is used for draining blast coils or very large direct radiating units. It is designed for a pressure of 10 pounds per sq. in. or less.

For typical applications of Dunham traps to various conditions of service, see details on pages 226-a and -b.

Monash-Younker Radiator Return Valves. The *Monash-Younker Co.* manufactures a float trap known as a *Radifier* (Figs. 47 and 48) for use on vacuum heating systems. The sectional view shows a hollow float immersed in a special float chamber, and so arranged with ports at top and bottom as to permit the passage of air through the float at all times when the vacuum

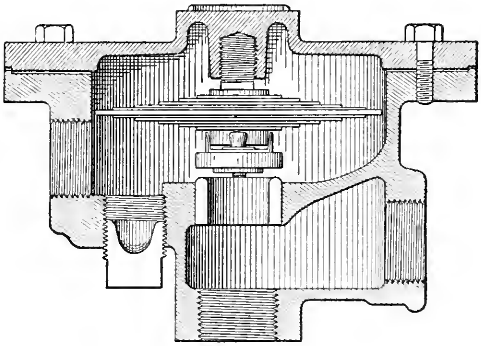


FIG. 46. DUNHAM BLAST TRAP.

pump is pulling on the system sufficiently to overcome the small head existing in the float chamber. Whenever an accumulation of water occurs sufficient to flood the float chamber, the float is raised from its seat and the water is discharged directly to the return.

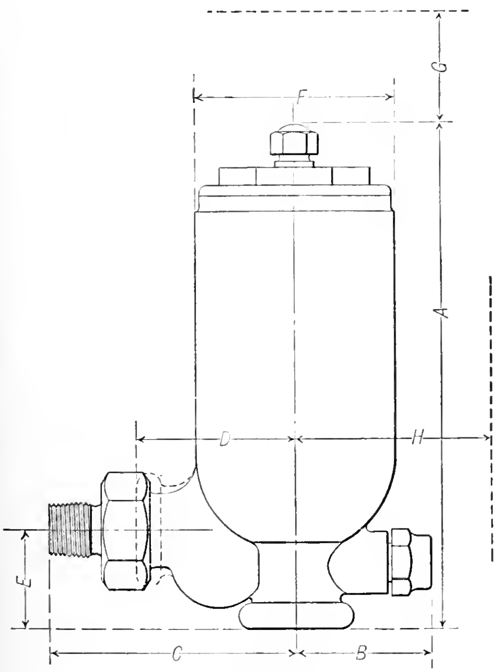


FIG. 47.

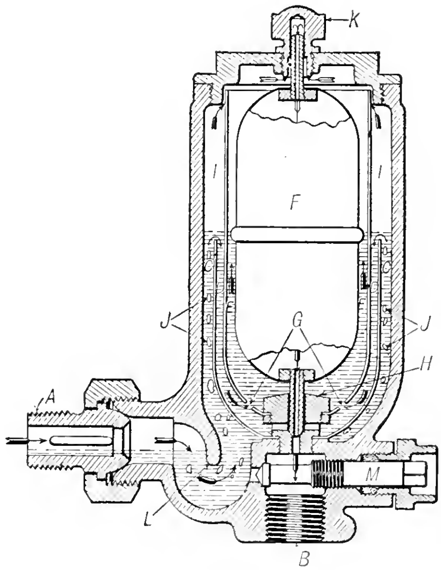


FIG. 48.

MONASH RADIFIER VALVE.

The by-pass plug M is provided for blowing out sediment deposited in the dirt pocket at L. Connection is made to the return hub of the radiator by means of the union at A.

The dimensions and capacities of these valves are given in the following tables:

TABLE 16

DIMENSIONS OF MONASH NOISELESS RADIFIER, CLASS B

(See Fig. 47)

	Number of Valve	10	11	12	13	14	15
A	Height, Inches	5¼	6 3/16	7¾	9⅞	11	12 5/16
B	Center to End of Lockshield	1⅜	2 1/16	2 1/16	2⅜	2 7/16	2 7/16
C	Inlet End to Center of Outlet	3 7/16	3⅞	3⅞	...	...	...
D	Inlet End to Center of Outlet	...	...	...	2⅝	3	3
E	Inlet Center to Outlet End	1½	1½	1½	1 11/16	2	2
F	Diameter of Body, Inches	2 7/16	2 15/16	3 1/16	3¾	4 9/16	5 13/16
G	Space to be Allowed for Removing Float	2½	3⅛	3¼	4⅛	4⅜	4¾
H	Space to be Allowed for Removing By-Pass Stem	3¼	4	4	4⅞	5 3/16	5 3/16

See note Table 15

TABLE 17

CAPACITY AND SIZE OF MONASH NOISELESS RADIFIER, CLASS B

No.	Pounds Water per Hour	Square Feet Radiation, Cast Iron	Wall Coils		Ceiling Coils		Size Pipe Connection, Inches	Weight of Radifier Ready for Shipment, Lb.
			Square Feet	Lineal Feet 1-inch Pipe	Square Feet	Lineal Feet 1-inch Pipe		
10	10	40	30	80	24	66	½	3¼
11	17	65	40	108	36	104	½	4
12	32	125	70	203	56	163	½	5¼
13	76	300	170	494	132	383	½	9¾
14	150	600	340	987	275	799	¾	14¾
15	300	1200	676	1963	528	1534	¾	22

NOTES.—Sizes No. 10, 11, and 12 are made with Union Coupling Connection at inlet. Sizes No. 13, 14, and 15 are made with Tapped Connection at both inlet and outlet.

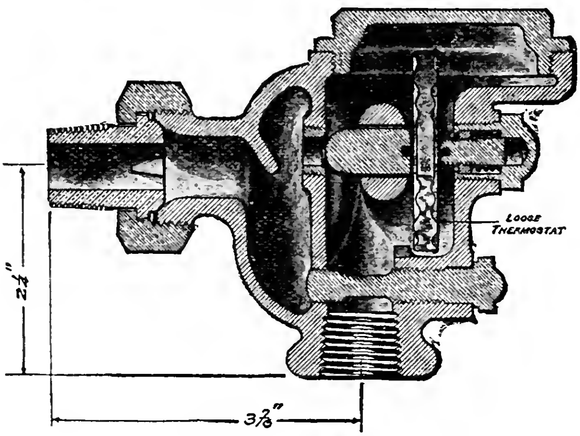


FIG. 49. MONASH THERMOSTATIC VALVE.

The *Monash thermostatic valve* (Fig. 49) has the thermostatic element placed on the discharge side of the trap and it is entirely free even when in position in the body of the trap. The vertical seat is protected with a deflector and dirt pocket in front of same, and a by-pass plug provided.

The spherical end of the plug-shaped disc is centered with respect to the discharge port and compelled to move perpendicularly to the seat by fixed guides. Excessive pressure is taken up by a spring plug back of the loose thermostatic element, so that should this element be overheated a secondary expansion of the latter may take place after the valve disc has been driven to its seat; this spring plug providing for additional expansion.

The $\frac{1}{2}''$ valve has a capacity of 300 sq. ft. of direct steam radiation.

Webster Radiator Return Valves. *Warren Webster & Co.* makes three types of thermostatic vacuum return valves; the first is of the expansion post type, the second of the float type, and the third of the expansion chamber with volatile liquid type.

The *Webster thermostatic* or *thermo valve* (Fig. 50) is used principally on small units of radiation, is adjustable and automatic, permitting water and air to pass, but expanding when the plug is surrounded by steam, causing it to seat and preventing waste of steam to the return.

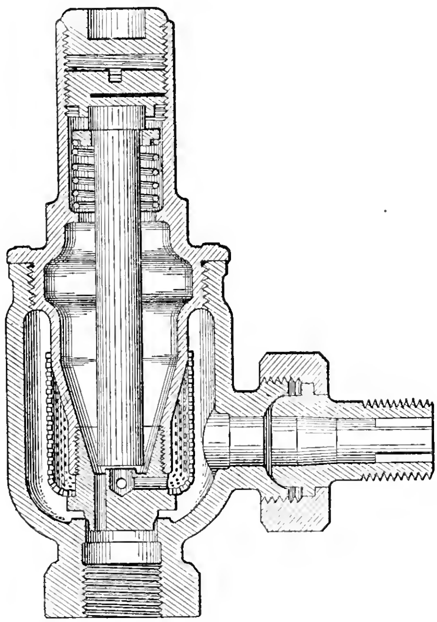


FIG. 50. WEBSTER THERMO VALVE.

This valve is intended for fairly constant low pressures. It is made in $\frac{1}{2}''$ size only, but connections $\frac{3}{4}''$ size may be obtained for special conditions.

There are four holes each way in the seat of the valve about $\frac{1}{8}''$ in diameter, as shown in the cut running both horizontally and vertically.

The *Webster Water Seal Motor* (Fig. 51) is all metallic, automatic, non-adjustable, and works on the float principle. Air passes out around the screw thread spindle. Water surrounding the float causes it to rise, opening the port and allowing the water to pass to the return pipe. This valve is intended for low-pressure service. Any vapors which may pass to the returns are readily condensed in these bare pipes.

These valves are made in $\frac{1}{2}''$, $\frac{3}{4}''$, $1''$ and $1\frac{1}{4}''$ sizes.

The port opening in the seat is $^5/_{32}''$ in diameter, and the bore of the tube in float is $^9/_{32}''$ in diameter.

The *Webster Sylphon Thermotor* (Fig. 52) is thermostatic in principle, operating under the

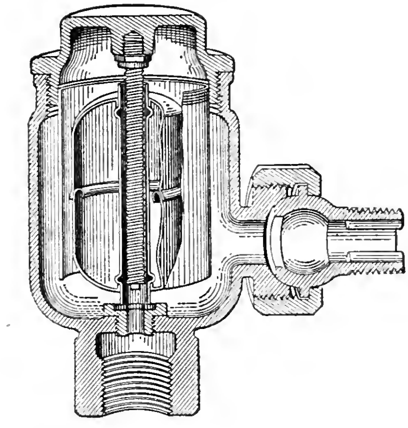


FIG. 51. WEBSTER WATER SEAL MOTOR.

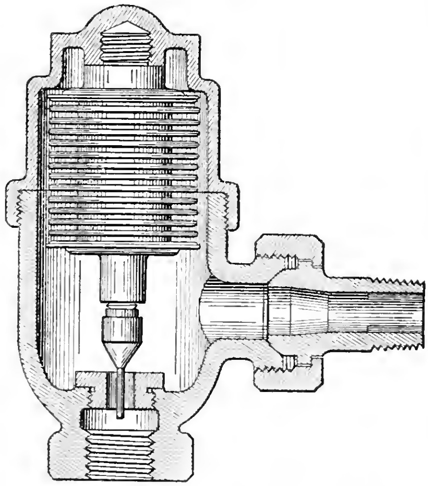


FIG. 52. WEBSTER SYLPHON THERMOTOR.

influence of an expansion bellows which contains a volatile liquid. The numerous corrugations in this bellows allow for a large lift of the valve from the seat, thus permitting dirt that would lodge in most valves to pass through the port. The valve is tested and adjusted in the factory once for all, and is good for widely varying pressures, permitting water and air to pass while tight against steam leakage. It is made in $\frac{1}{2}''$, $\frac{3}{4}''$, $1''$ and $1\frac{1}{4}''$ sizes.

The port in the seat is $^5/_{16}''$ in diameter and the lift of the valve about $\frac{3}{8}''$.

A *high pressure modification* of the Sylphon Thermotor has been designed and adapted to varying pressures up to about 90 pounds gage.

TESTS OF RADIATOR RETURN TRAPS

Radiator traps are often tested in order to determine their effectiveness in preventing the passage of steam, as well as their ability to discharge air and water readily.

The test data available are rather limited, and vary more or less with the manner in which each investigator has arranged his apparatus. It is probably true, however, that the results from any one series of tests furnish data sufficiently accurate for purposes of comparison.

James A. Donnelly and *J. W. Hook* have conducted extensive tests on a large number of commercial types of radiator traps, each investigator using a different method of testing.

Donnelly's tests are in accordance with the method accepted by the *Amer. Soc. of Heat. and Vent. Engrs.*, Jan., 1913.

In these tests the object was to determine the degree of vacuum (Table 18) which could be maintained on the returns with the vacuum pump running at constant speed, when various radiator traps were installed on the same radiator, and connected to the same return system with no injection water supplied for condensing the steam, if any, passed by the trap.

"This apparatus consisted of a receiver into which low pressure steam was admitted and which was suitably drained from the bottom. Steam was then taken from the top of the receiver and controlled in its admission to the radiator by a fractional inlet valve. A thermometer is provided in the steam inlet to the radiator as well as in the outlet, immediately preceding the vacuum system valve. A short section of $\frac{3}{4}''$ and $1''$ pipe leads to the vacuum pump, which is a Bishop-Babcock-Becker No. 121, duplex single-acting cylinders, $2\frac{1}{4}''$ in diameter by $3''$ stroke, running at 100 strokes per minute. Upon the radiator a pressure gage is provided and upon the return line a vacuum gage.

"The radiator was tested for condensation without the return piping and it was found to condense an average of 8 lbs. 10 oz. per hour at 142 degrees difference. This amount of condensation multiplied by 970 and divided by 250 makes the radiator equivalent to 33.46 sq. ft. of direct surface. Dry steam from the receiver was connected to the return piping, the pump was operated at its usual speed, and atmospheric pressure maintained. The return piping and pump with 142 degrees difference gave a condensation which averaged 1 lb. 6 oz. per hour. This multiplied by 970 gives 1334 B.t.u. as given off by the return piping, divided by 250 this makes the return piping and radiating and pumping effect of the vacuum pump as equal to 5.33 sq. ft. of standard surface. Dividing the surface of the return piping by the surface of the radiator gives 16% as the proportionate radiating effect of the return piping in comparison with the radiator. This proportion is somewhat higher than would usually be met with in ordinary plants and the apparatus will therefore produce a somewhat better vacuum with all valves than would ordinarily be obtained.

TABLE 18

TESTS OF VACUUM SYSTEM VALVES

(James A. Donnelly)

Device	Vacuum Maintained, inches of mercury
1/32 inch orifice	12
3/64 " "	9
1/16 " "	6
Rubber thermostatic valve 1/12 turn open	5
Monash Radifier	2.5
Illinois Engineering-float	2.5
Water Seal Motor	3.0
D. G. C. Radial Disc	4.5
Consolidated Engineering-float	7.0
No. 1 Haines Vaporizing Fluid	5.0
Illinois Engineering Vaporizing Fluid	9.0
No. 2 Haines Vaporizing Fluid	16.0
Willwork Vaporizing Fluid	10.0
Sylphon Vaporizing Fluid	18.0
Kinealy Vaporizing Fluid	19.0
Bishop-Babcock Vaporizing Fluid	19.0
Thermograde Vaporizing Fluid	19.0
Differential Valve	5 to 19.0 (as weighted)
Dunham Vaporizing Fluid	19.0

"Any tendency of the valves to retain water or air in the radiator was immediately detected by a drop in the temperature at the outlet of the radiator. This occurred with some of the vaporizing fluid valves, and in obtaining the proper adjustment of rubber thermostatic and needle valves, but the valves were easily adjusted and the trouble remedied.

"This resulted in a temperature at the outlet of the radiator, which as nearly as can be observed, was substantially the same as that at the inlet. The vacuum pressures maintained upon the return line with the various valves and devices were not necessarily those that would be produced by the use of these valves, but they show the best result that would be possible, provided all the valves were in first-class operative condition and free from any dirt which might hold them open and permit the escape of steam."

Hook's tests were run with a constant vacuum on the return mains, and the amount of steam passed by the trap determined by condensing the same in a suitable condenser placed just beyond the trap in the line to the pump. In general, the trap passing the least steam for a given vacuum, provided it does not hold back water or air in the radiator, is most satisfactory. In such a series of comparative tests a correction is always necessary, due to the fact that some of the condensate passing the trap will "flash" into steam, or re-evaporate due to the lower pressure on the return side of the trap.

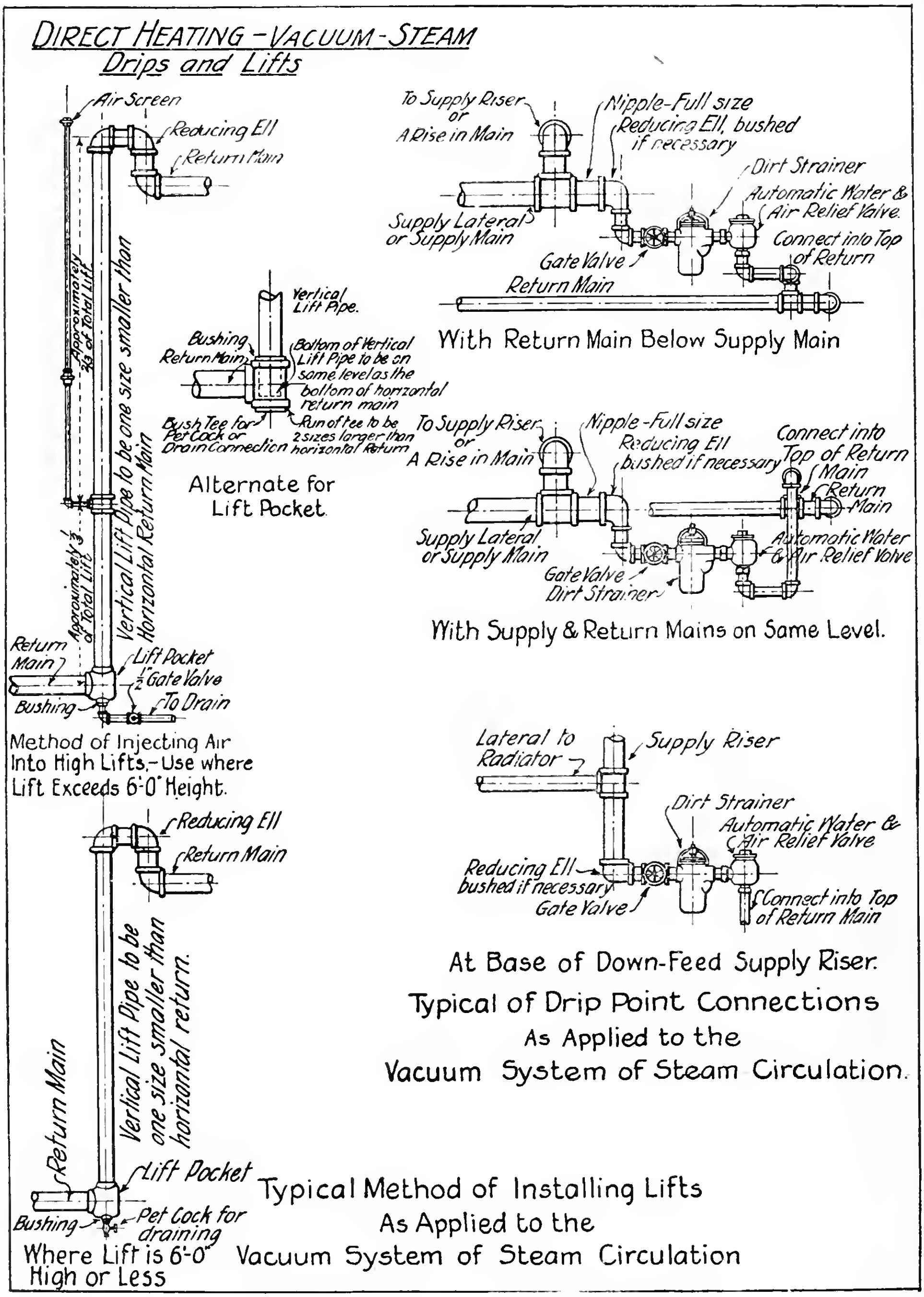
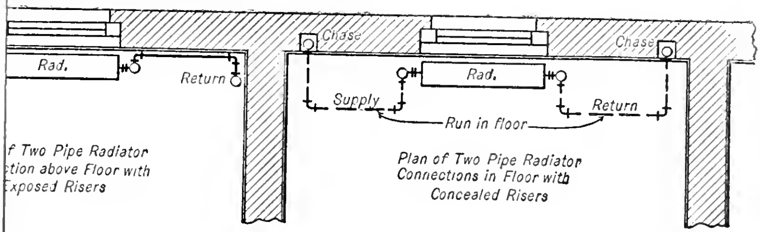
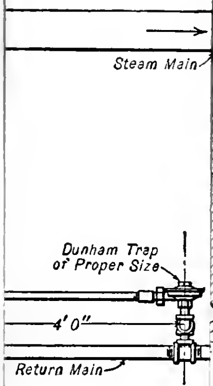
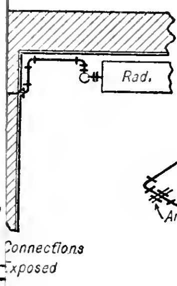
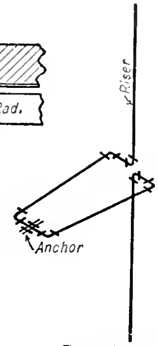


FIG. 53. RETURN CONNECTIONS FOR WEBSTER VACUUM SYSTEM.

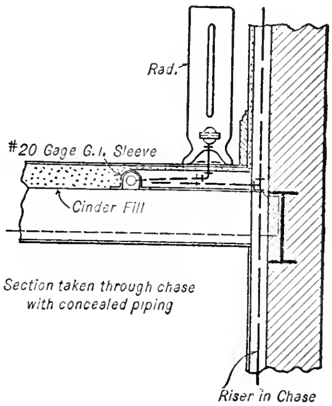






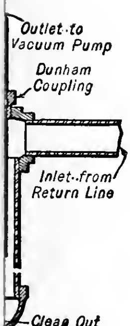


Expansion Loop on risers every 5 or 6 stories

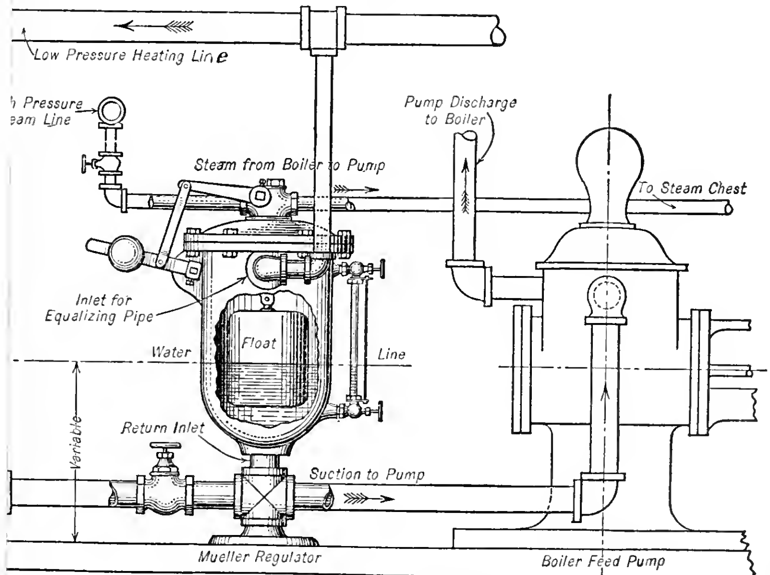


Section taken through chase with concealed piping

OWING METHOD OF
IN STEAM MAIN.



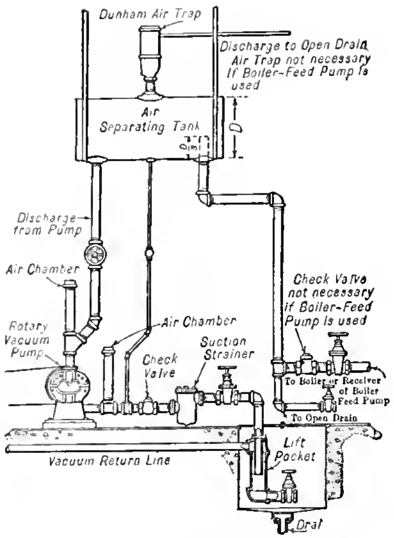
E, SHOWING CON-
LIFT IN A VACUUM



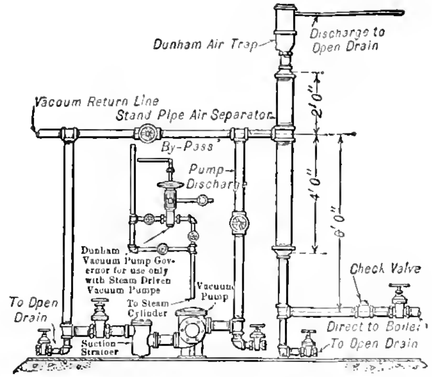
No.	Draining Capacity, Lineal Feet Radiation	Size Steam Valve	Size Inlet and Outlet Connections
0	5,000	½	1¼
1	10,000	¾	2
2	20,000	1	2½
3	30,000	1¼	3
4	50,000	1½	4

ONDENSATION RECEIVER AND AUTOMATIC PUMP REGULATOR.

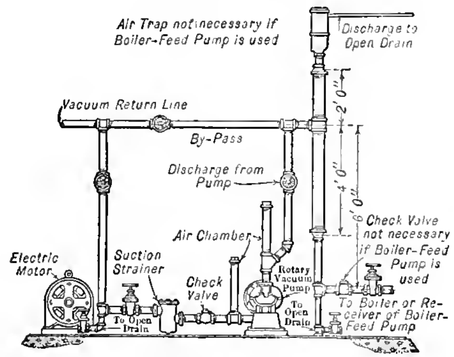
-Water line of receiver should be at least 3′ 0″ above center line of pumps.



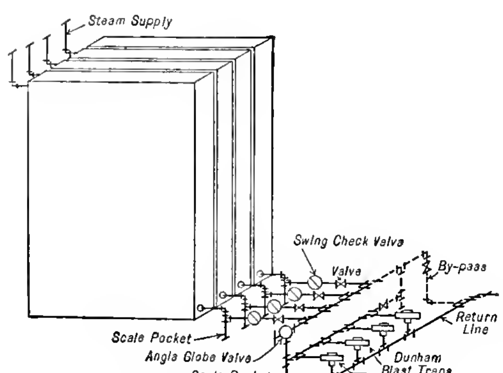
DETAIL NO. G, SHOWING METHOD OF CONNECTING ROTARY VACUUM PUMP WHEN RETURN LINE IS BELOW VACUUM PUMP.



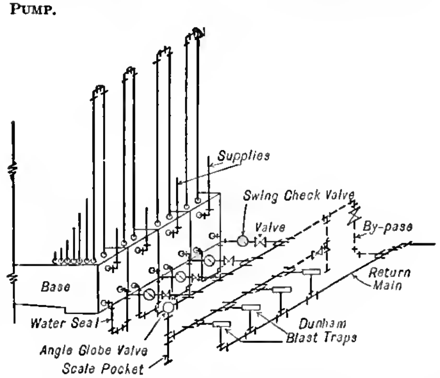
DETAIL NO. D, SHOWING METHOD OF DISCHARGING DIRECT FROM VACUUM PUMP INTO BOILER.



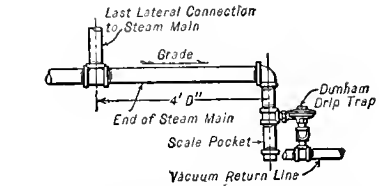
DETAIL NO. F, SHOWING METHOD OF CONNECTING UP ROTARY VACUUM PUMP WHEN RETURN IS ABOVE PUMP.



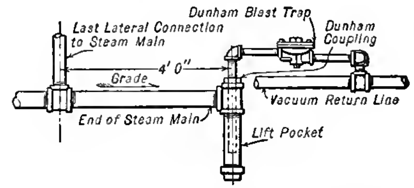
DETAIL NO. I, SHOWING METHOD OF APPLYING DUNHAM BLAST TRAPS TO BLAST HEATERS.



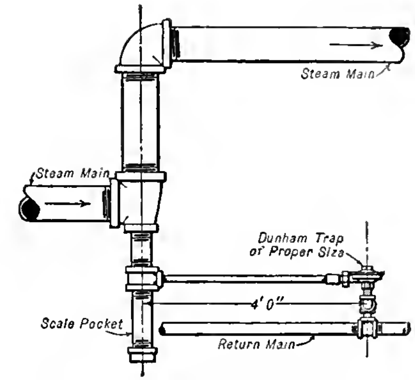
DETAIL NO. J, SHOWING METHOD OF APPLYING DUNHAM BLAST TRAPS TO PIPE COIL HEATERS



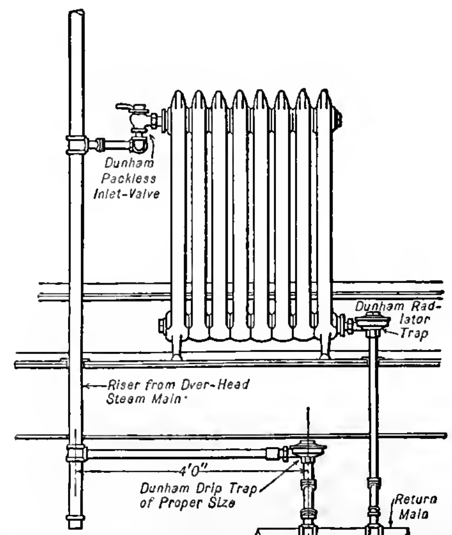
DETAIL NO. A, SHOWING METHOD OF DRAINING END OF STEAM MAIN WHEN RETURN LINE IS BELOW STEAM MAIN.



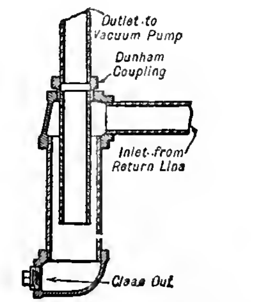
DETAIL NO. B, SHOWING METHOD OF DRAINING END OF STEAM MAIN WHEN RETURN LINE IS ABOVE STEAM MAIN. Note construction of the Lift Pocket (see Detail No. G).



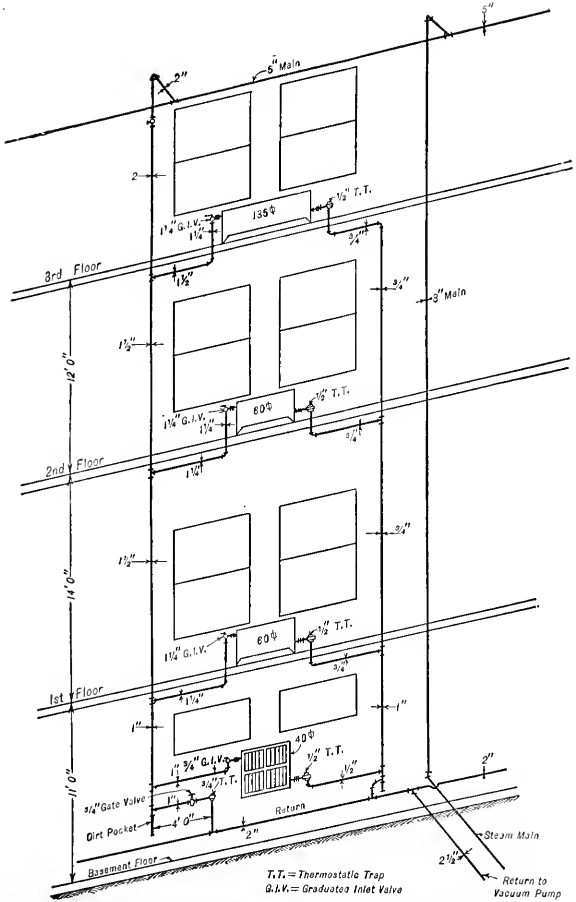
DETAIL NO. H, SHOWING METHOD OF DRIPPING RISE IN STEAM MAIN.



DETAIL NO. C, SHOWING METHOD OF DRAINING BOTTOM OF STEAM RISER IN AN OVERHEAD SYSTEM.

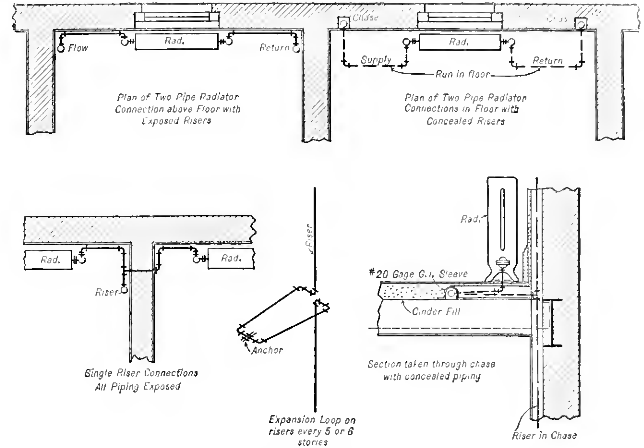


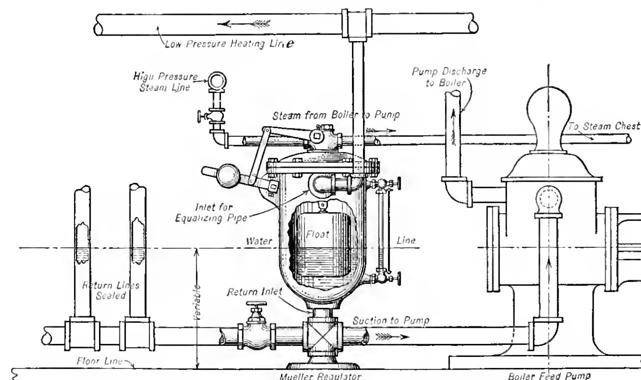
DETAIL NO. E, SHOWING CONSTRUCTION OF LIFT IN A VACUUM RETURN LINE.



Riser Diagram for Down-Feed Mechanical Vacuum System.

Note.—Pipe sizes proportioned from Table 24, steam main from col. A, return main from col. C, and branch returns from col. E.





No.	Draining Capacity, Lineal Feet Radiation	Size Steam Valve	Size Inlet and Outlet Connections
0	5,000	½	1¼
1	10,000	¾	2
2	20,000	1	2½
3	30,000	1¼	3
4	50,000	1½	4

Condensation Receiver and Automatic Pump Regulator.

Note.—Water line of receiver should be at least 3′ 0″ above center line of pumps.

This re-evaporation must take place at the expense of the heat of the liquid, and is dependent on the difference between the heat of the liquid at the pressure existing in the last section of the radiator and the heat of the liquid at the pressure of the returns.

Thus the re-evaporation per pound is:

$$R = \frac{q_1 - q_2}{r_2} \times W, \text{ in which}$$

q_1 and q_2 = heat of liquid in the radiator and the returns. r_2 = heat of vaporization in the returns.

W = weight of water passed by trap as measured.
G = " " steam " " " " "
Then $G - R$ = corrected weight of steam passed by trap during test.

A second solution is really necessary, since W is not all of the water passing the trap, as it does not include R, which vaporizes instantly on passing into the returns. In the second solution use $W' = W + R$ as obtained in the first solution. The error is small.

The test results are reported at length in a thesis by *Mr. Hook* and show about the same variation as indicated in Table 18.

PIPING DETAILS AND CONNECTIONS FOR DIRECT STEAM HEATING

In the actual installation of most steam-heating systems difficulties and obstructions of various kinds must be overcome without interfering with the proper operation of the system installed. Furthermore, drainage and expansion must be provided for, and the mains must be suitably connected to the boiler for the proper outflow of dry steam and the prompt return of condensation.

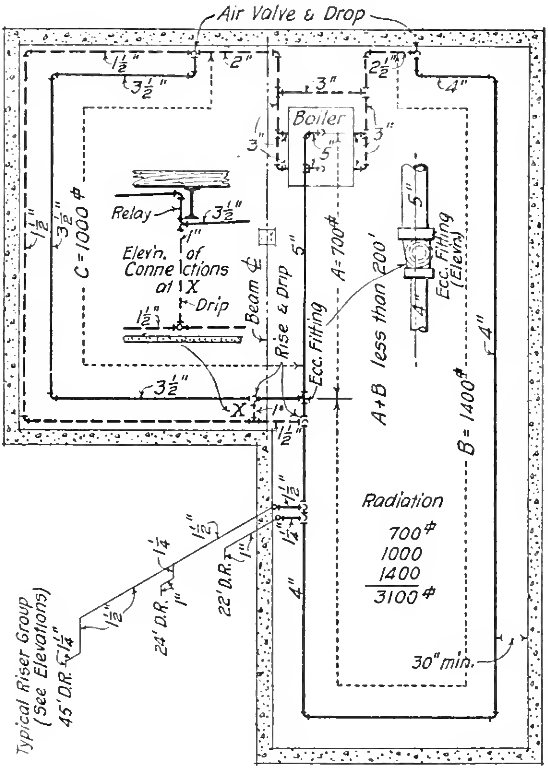
Mains and Branches. Mains and main branches must be *uniformly* graded as shown in Figs. 1 and 3, allowing for a fall of about 1″ to ¾″ in 10′–0″ for the steam mains and 1″ in 30′–0″ for the wet or dry return mains. Where mains must pass under beams or be "relayed" to secure head room, suitable drips must be installed to drain all pockets of condensation. See Fig. 54 and elevation at X.

Whenever it becomes necessary to reduce the size of a steam main, provision should be made for free flow of any condensation in the piping by installing an "eccentric" fitting as shown in Fig. 54, where the 5″ main serves a 4″ and 3½″ branch. The run of the tee is made eccentric so that the bottom of the 5″ and 4″ lines is at the same level, thus preventing the fitting from holding back any condensation.

Risers. Risers may drain back directly into the steam main in one-pipe work or be dripped, as shown in Figs. 6 and 8 to a separate return. In the former case the branch supplying the riser is so connected as to also drip the main. The *horizontal branch* supplying the riser must be of sufficient length, usually not less than 30″, to permit of some movement by the main and the riser. The mains therefore must be kept from 2′–0″ to 3′–0″ from the basement walls if the radiators are to be placed along outside walls on the upper floors. See Fig. 54 for a typical layout of a one-pipe system. In two-pipe work the supply risers are not dripped as they carry no radiator condensation. Very long risers should be provided with swing expansion joints.

Radiator Branches. Radiator branches on upper floors must be so run that suitable allowance will exist for the vertical expansion of the riser to which they are connected. This expansion is most easily provided for by making up the branch with double elbows, which allows of a slight swinging movement taken care of by the turning of the nipples in the elbows.

These radiator branches are preferably run in the floor construction, where they would be concealed from view. In the case of floors with cinder fill suitable conduits may be formed of No. 18 or 20 U. S. S. gage galvanized iron bent into an inverted U-section. If this cannot



Basement Plan - One Pipe Steam System

Recommended returns = ½ steam main + 1 size

MAIN TABLE

Steam Main	Return Dry	Return Wet	Radiation Sq. ft.
1"	¾"	¾"	40
1¼	1	¾	75
1½	1	1	126
2	1¼	1	286
2½	1½	1	535
3	1½	1	890
3½	2	1	1360
4	2	1	1950
5	2½	1¼	3600
6	3	1½	5900
8	4	2	12700
10	5	2½	22900
12	6	3	37000

Runs up to 200'

For longer runs reduce above rad. values as follows: $R \times \sqrt{\frac{200}{L}}$

Measure all runs back to boiler

RELIEFS (One Pipe Relief System)

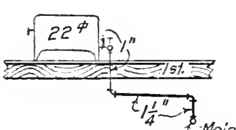
Riser	1"	1¼	1½	2	2½	3	3½	4	4½	5
Relief	¾"	1	1	1¼	1½	2	2	2½	3	3

A.C.W.

FIG. 54.

MAIN SIZES TABLE 19.
RELIEF SIZES TABLE 20.

ONE PIPE STEAM HEATING PIPING SIZES



First Floor Radiator

FIG. 55.

BRANCH AND RISER TABLE

Square feet Radiation		Rad Tapping	Branch Riser Rad Arm
0	20	1	1
21	24	1	1¼
25	40	1¼	1¼
41	60	1¼	1½
61	80	1½	1½
81	100	1½	2
101	200	2	2

N.S.T.

TABLE 21.

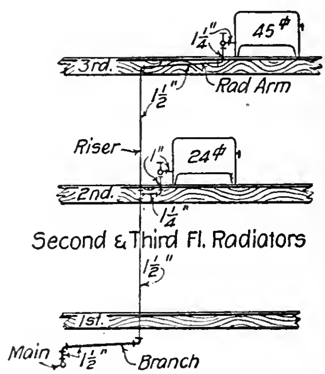


FIG. 56.

be done they should be run at the ceiling below, and brought up through the floor directly at the radiator. In no case should they be run above the floor for a distance greater than 8″ or 9″, and the union sleeve on the radiator valve should always screw directly into the radiator tapping or bushing.

Typical return and drip connections for vacuum systems are illustrated in Fig. 53 and are adapted from the present practice of the *Warren Webster Co.*

PIPING SIZES FOR DIRECT STEAM HEATING.

The sizes of mains, branches, risers and radiator connections for steam heating depend primarily on *the allowable friction pressure loss* between the boiler and the last radiator in the system. In addition to this limitation the boiler outlets must be large enough so that the outlet velocity through same will not be sufficient to lift water into the mains, and in one-pipe systems the steam flowing up the risers must not carry back the down-coming condensation. These two latter conditions seem to be satisfied with velocities ranging from 12′ to 16′ per second.

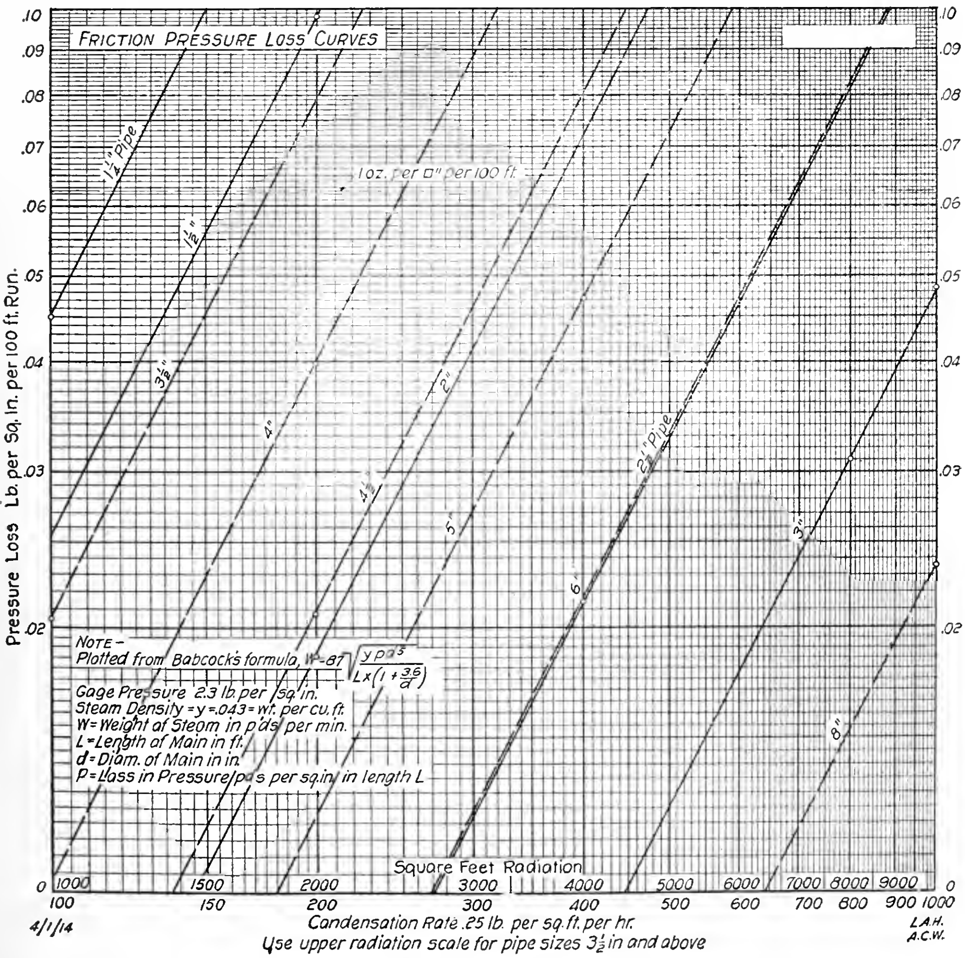


FIG. 57.

Piping Sizes Based on Velocity. It would of course be possible to lay out a piping system by assuming *velocities*, since the maximum quantity of steam required for any part of the system can be easily calculated from the sq. ft. of radiation installed. Hence, if Q = the volume of steam in cu. ft. per sec., we would have $A = Q/V$, where V = the velocity in ft. per sec. and A = the area of the pipe in sq. ft.

Piping Sizes Based on Pressure Drop. In the best practice it is generally customary to assume an allowable pressure drop per 100′ of main as the basis for proportioning the piping sizes. This drop is usually taken at 1 oz. per 100′ of straight piping or its equivalent, for installations within separate buildings. The common tables of pipe sizes for low-pressure steam heating are based on this drop, but in the case of groups of buildings or wherever there are excessively long runs it may be advisable to allow a greater drop in order to decrease the pipe size on such runs.

Babock's formula for the pressure drop due to steam flowing in a line of known diameter has already been given in the chapter on "Steam," and may be put in the form:

$$p_1 - p_2 = L \times W^2 \times \frac{1}{7569y} \times \left(\frac{1 + \frac{3.6}{d}}{d^5} \right),$$

where p_1 is the initial and p_2 the final pressure in pounds per sq. in., L = the total equivalent length in feet, W = the weight of steam flowing in pounds per min., y = the density or weight of steam per cu. ft. in pounds at pressure p_1, and d = nominal diameter of pipe in inches.

The logarithmic chart shown in Fig. 57 was constructed by the use of this formula, and may be used for proportioning piping for low-pressure heating systems for any drop in pressure less than 0.1 lb. per sq. in. per 100′ of run.

Elevation of Water in Returns. A further application of the *Babcock* formula showing the practical limitations placed upon a gravity steam heating system, unless the piping has been de-

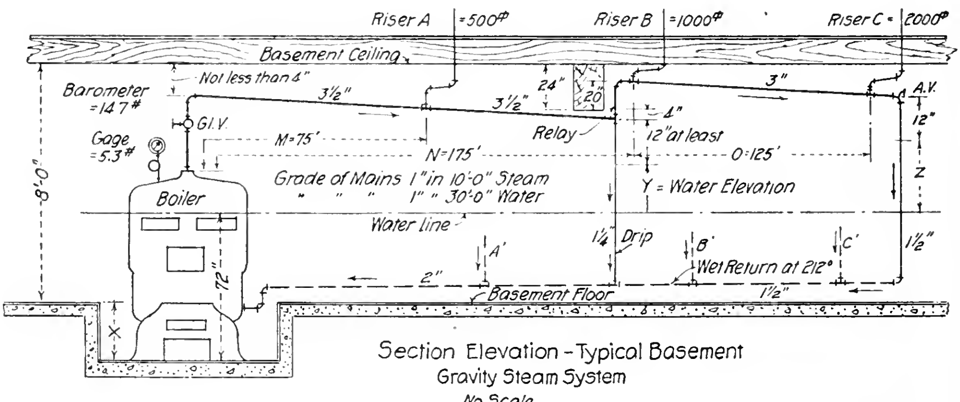


FIG. 58.

signed for a reasonable friction pressure loss, is shown in Fig. 58, where, due to small mains and an excessive drop in pressure, the corresponding elevation of the water in the returns requires a deep boiler pit to prevent flooding the far end of the steam main. This elevation of water in the returns of a closed gravity circulating system is due to the fact that as steam flows along the supply main it is losing pressure due to friction, and hence the steam pressure at the end of the

main is always less than that of the steam in the boiler. Since the return main connects this point of low pressure directly with the boiler, and since the velocity of flow of water in the returns is very small, as its volume is only 1/1700 of the volume of the steam from which it is condensed, the water will rise in the drop at the end of the main until it establishes a static head equal to the pressure difference between the boiler and the end of the steam main.

Moreover, this head is not negligible, for, with a difference of one pound between the boiler and the end of the main, water at 212° F., weighing 59.76 lb. per cu. ft., will rise until this head is $1 \Big/ \frac{59.76}{144} \times 12 = 28.92''$. In addition to the allowance for the elevation of the water in the drops the depth of pit may have to be further increased to provide for uniform grade of mains and the avoidance of beams or girders, as a reference to the figure will show.

Example. A *calculation for depth of boiler pit* under conditions noted above and as given in Fig. 58 may be made as follows, using *Babcock's* formula in the form just stated. It is assumed that we have a gravity circulating system and that the only load on the covered main is the steam radiation indicated on risers A, B, and C. Condensation in the main has been neglected.

First, find pressure loss between boiler and riser A. As all the steam flows through this section, and the entrance resistance at boiler outlet is taken equal to one globe valve, we have:

$$W = \frac{3500 \times 0.275}{60} = 16.1 \text{ lb. and } L = 75' + 2 \times 16.5' + 11' = 119 \text{ feet}$$

where 16.5′ and 11′ are the pipe equivalents for a 3½″ globe valve and elbow. One sq. ft. of the radiation installed condenses 0.275 lb. per hour. Then find the value of the expressions containing y and d, the former at 5.3 lb. gage and the latter for 3½″ pipe. Then $p_1 - p_2 = 119 \times (16.1)^2 \times 0.00265 \times 0.00381 = 0.316$ or 0.32 lb. per sq. in. and the pressure at foot of riser $A = 5.3 - 0.32 = 4.98$ lb.

Second, for the pressure loss between risers A and B we have $W = \dfrac{3000 \times 0.275}{60} = 13.75$ lb. and $L = 102 + 2 \times 11 = 124$ feet, and the term including y must be found at 4.98 lb., while d is the same as before, $= 3½''$.

Then $p_1 - p_2 = 124 \times (13.75)^2 \times 0.00271 \times 0.00381 = 0.244$ or 0.24 lb. per sq. in. and the pressure at foot of riser $B = 5.3 - (0.32 + 0.24) = 4.74$ lb.

Third, for the pressure loss between risers B and C we have $W = \dfrac{2000 \times 0.275}{60} = 9.17$ lb. and $L = 125 + 0 = 125$ feet, and the term including y must be found at 4.74 lb. while $d = 3''$. Then $p_1 - p_2 = 125 \times (9.17)^2 \times 0.00273 \times 0.00905 = 0.259$ or 0.26 lb. per sq. in., and the pressure at foot of riser $C = 53 - (0.32 + 0.24 + 0.26) = 5.3 - 0.82 = 4.48$ lb.

Now 1 foot head of water per sq. in. at 212° weighs $\dfrac{59.76}{144}$ lb., and hence the head in inches equivalent to 1 lb. $= \dfrac{144}{59.76} \times 12 = 28.92''$ and the elevation at Y* to offset the pressure loss $= 0.56 \times 28.92'' = 16.2''$ head, and at $Z = 0.82 \times 28.92 = 23.7''$ head of water.

The depth of pit must provide not only for the maximum elevation of water in the drops, but also for the clearance required for passing the beam and grade of main, plus all other clearances as indicated in Fig. 58.

* **Note.**—The actual elevation of water at Y will be somewhat less than shown here, as the pressure loss at the drip connection is not quite as great as at riser B by the two fittings and pipe between Y and B.

Consider the two conditions at Y and at Z, using a boiler with a 72″ water line.

At the "relay," Y		At the end of main, Z	
Beam	= 24″	Clearance of relay	= 4″
To center of tee	= 4	Grade of pipe	= 12.5
Allowance	= 12	Allowance	= 12
Elevation at Y	= 16.2	Elevation at Z	= 23.7
Water line	= 72	Water line	= 72
	128.2		124.2
Story height	= 96.	Story height	= 96.
Pit = X	= 32.2	Pit = X	= 28.2

It will be seen that although the water elevation is greatest at Z, yet the elevation at Y combined with the depression caused by the beam requires a deeper pit. Practically, a pit 36″ deep should be used, or preferably mains of proper size should be used which by Chart (Fig. 57) at 3500 sq. ft. should be 4½″ diameter for first section.

Tables of Pipe Sizes for Steam Heating. Piping sizes for steam-heating systems may be calculated from the above formula, but are ordinarily *tabulated* to read in square feet of direct radiation supplied rather than in pounds of steam. These tables assume that each sq. ft. of direct radiation condenses 0.25 lb. of steam per hour, or that the radiator contains steam at 2 lb. pressure and stands in still air at 70° F.

Sizes for *One-Pipe Gravity Systems* as used by the *U. S. Treasury Dept.* are given for steam and return mains by Table 19 and for branches, risers, and tappings by Table 21, and their appli-

TABLE 22

SIZES AND CAPACITIES OF MAINS, BRANCHES, AND RISERS FOR TWO-PIPE STEAM-HEATING SYSTEMS—GRAVITY CIRCULATION—MAINS 200′ IN LENGTH*

(Alfred R. Wolff)

	For Two Pounds and Five Pounds Steam Pressure				
1	2	3	4	5	6
Diameter of Supply in Inches	Diameter of Return in Inches	Two Pounds Pressure Total Heat Transmitted in B.t.u. per Hour	Radiating Surface in Square Feet	Five Pounds Pressure Total Heat Transmitted in B.t.u. per Hour	Radiating Surface Square Feet
¾	¾	5,000	20	10,000	40
1	¾	9,000	36	15,000	60
1¼	1	18,000	72	30,000	120
1½	1¼	30,000	120	50,000	200
2	1½	70,000	280	120,000	480
2½	2	132,000	529	220,000	880
3	2½	225,000	900	375,000	1,500
3½	2½	330,000	1,320	550,000	2,200
4	3	480,000	1,920	800,000	3,200
4½	3	690,000	2,760	1,150,000	4,600
5	3½	930,000	3,720	1,550,000	6,200
6	3½	1,500,000	6,000	2,500,000	10,000
7	4	2,250,000	9,000	3,750,000	15,000
8	4	3,200,000	12,800	5,400,000	21,600
9	4½	4,450,000	17,800	7,500,000	30,000
10	5	5,800,000	23,200	9,750,000	39,000
12	6	9,250,000	31,000	15,500,000	62,000
14	7	13,500,000	54,000	23,000,000	92,000
16	8	19,000,000	76,000	32,500,000	130,000

*Note.—(1) Return sizes 1″ and above may safely be cut at least one-pipe size.
(2) Each sq. ft. radiation is assumed to transmit 250 B.t.u. per sq. ft. per hour.

Special Application.—Many designers use this table at 2 lb. for proportioning steam mains and branches in vacuum systems, together with Table 24, Columns C and E, for the return mains and branches. See page 226-c for example.

cation to practice illustrated in Figs. 54, 55 and 56. Table 20 applies to reliefs as shown in Fig. 6. In one-pipe work, unless main is dripped, it must run *full size to drop*, at which point an *automatic air valve* should be placed.

Sizes for *Two-Pipe Gravity Systems* are given in Table 22, and their application is similar to that shown for one-pipe work except that the size of steam main may be reduced as rapidly as withdrawal of steam will permit.

In the original table *A. R. Wolff* limited the use of this table to mains 100′ in length, but *N. S. Thompson*, of the *U. S. Treasury Dept.*, has found that the capacities are ample for runs 200′ long, including fittings, etc., measured *from the boiler outlet*. For *longer runs* multiply capacities given in columns 3, 4, 5, and 6 by $\sqrt{\frac{200}{L}}$, where L = length in ft.; as for example a 4″ main 400′ long will supply $\sqrt{\frac{200}{400}} \times 1920$ sq. ft. at 2 lb. pressure, or $0.71 \times 1920 = 1360$ sq. ft. See Fig. 7 for type of system to which the above table applies.

Sizes for *Vacuum Systems*, where not over 2 pounds pressure is available at entrance to heating main, are given in Table 23. In this case much smaller returns* may be used than in straight gravity work and the condensation may be actually lifted by the vacuum pump through small heads. Typical return connections for drips and lifts in vacuum work as used by the *Warren Webster Co.* are shown and explained in Fig. 53. It should be noted that if the condensation is to be lifted a positive seal must be used at the point where the lift is to be made.

TABLE 23

DIRECT HEATING—PIPE SIZES FOR VACUUM SYSTEMS*

PIPE SIZES FOR GIVEN LOSS AND RUN

Loss in Pounds	Flow	1½	2	2½	3	3½	4	5	6	8	10	12
	Ret	¾	1	1¼	1½	1½	1½	2	2½	3	4	5
	Run	Radiation Square Feet Supplied										
⅛	100	194	427	745	1350	1930	2700	4800	7725	15800	28100	45600
¼	100	277	609	1060	1925	2750	3850	6840	11000	22500	40000	63000
½	100	388	854	1490	2700	3860	5400	9600	15450	31600	56200	91200
1	100	554	1218	2120	3850	5500	7700	13680	22000	45000	80000	126000
⅛	200	137	316	510	950	1370	1900	3410	5460	11250	20400	31600
¼	200	195	450	725	1350	1950	2725	4850	7775	16000	29000	45000
½	200	274	632	1020	1900	2740	3800	6820	10920	22500	40800	63200
1	200	390	900	1450	2700	3900	5450	9700	15550	32000	58000	90000
⅛	500	86	207	338	587	880	1210	2170	3400	7020	12650	20000
¼	500	123	295	430	850	1250	1725	3090	4850	10000	18000	28400
½	500	172	414	676	1174	1760	2420	4340	6800	14040	25300	40000
1	500	246	590	960	1700	2500	3450	6180	9700	20000	36000	56800
⅛	750	71	175	282	482	702	985	1755	2825	5700	10250	16150
¼	750	101	250	400	700	1000	1400	2500	4025	8100	14600	23000
½	750	142	350	564	964	1404	1970	3510	5650	11400	20500	32300
1	750	202	500	800	1400	2000	2800	5000	8050	16200	29200	46000
⅛	1000	61	147	240	440	632	880	1560	2500	4775	8950	13900
¼	1000	87	210	340	625	900	1250	2225	3600	6800	12700	19700
½	1000	122	294	480	880	1264	1760	3120	5060	9550	17900	27800
1	1000	174	420	680	1250	1800	2500	4450	7200	13660	25400	39400
⅛	1500	51	123	193	350	510	720	1280	2040	4220	7375	11800
¼	1500	72	175	275	500	725	1025	1825	2900	6000	10500	16800
½	1500	102	246	386	700	1020	1440	2560	4080	8440	14750	23600
1	1500	144	350	550	1000	1450	2050	3650	5800	12000	21000	33600
¼	2000	62	150	245	430	625	875	1600	2540	5100	9120	16200
1	2000	124	300	490	860	1250	1750	3210	5080	10200	18240	32400
¼	2500	54	133	211	383	555	775	1375	2225	4575	8071	12800
1	2500	108	266	422	766	1110	1550	2750	4450	9150	16142	25600
¼	3000	51	124	195	353	512	725	1280	2025	4400	7358	11500
1	3000	102	248	390	706	1024	1450	2560	4050	8800	14716	23000

* **NOTE.—In selecting the return sizes for vacuum systems use Table 24, Columns C and E, for mains and risers in single buildings. See page 226-c for typical application to vacuum work.**

Return Pipe Sizes for Direct Steam Heating. The sizes recommended for return pipes have already been indicated for each type of system for which the steam pipe sizes have been tabulated.

The *theoretical* size of the return main in any steam heating system will depend—first, on the allowable pressure drop in this main, usually taken the same as for the steam main; and, second, on whether this main is run *wet*, below the boiler water line, or *dry*, above the boiler water line.

In the case of "wet returns" the main handles only the water of condensation from radiators and steam piping, and, assuming the same friction pressure loss formulas apply to steam and water flowing in pipes, the weight of each will vary as the square roots of the densities for the same pressure drop and pipe size. Water at 212° F. = 59.76 lb. per cu. ft. and steam at 212° = 0.0370 lb. per cu. ft., so that the ratio $\sqrt{\frac{0.0375}{59.76}} = \sqrt{\frac{1}{1600}} = \frac{1}{40}$ or the capacity of a pipe as a wet return is 40 times its rating as a steam main, supplying steam at 212° F.

In the case of "dry returns" the return main not only carries condensation, but varying percentages of steam as well, depending on the type of system installed. The flow of this steam must be allowed for, as its presence is required to supply the radiation loss from the return main. Tables based on these varying percentages of steam which may be estimated from experience or taken from the notes appended to the tables are given below.

Table 24 by *James A. Donnelly* gives the maximum capacity of return mains, both for wet returns and for various percentages of steam carried in dry returns; as well as the steam rating for pipes from one-half to sixteen inches. These quantities are all figured for a drop in pressure of one ounce to one hundred feet in straight pipe.

TABLE 24

RETURN PIPE SIZES FOR DIRECT STEAM HEATING (*James A. Donnelly*)

Size in Inches	A	B	C	D	E	F	G	H
	Steam Rating Sq. Ft.	Wet Return A x 40	2½% Steam A x 20	5% Steam A x 13½	7½% Steam A x 10	10% Steam A x 8	15% Steam A x 5.7	20% Steam A x 4.4
½	5	200	100	67	50	40	27	22
¾	20	800	400	270	200	160	114	88
1	40	1,600	800	540	400	320	228	176
1¼	75	3,000	1,500	1,012	750	600	427	330
1½	150	6,000	3,000	2,024	1,500	1,200	855	660
2	300	12,000	6,000	4,050	3,000	2,400	1,710	1,320
2½	500	20,000	10,000	6,750	5,000	4,000	2,850	2,200
3	900	36,000	18,000	12,150	9,000	7,200	5,130	3,960
3½	1,500	60,000	30,000	20,250	15,000	12,000	8,550	6,600
4	2,000	80,000	40,000	27,000	20,000	16,000	11,400	8,800
4½	2,800	.....	56 000	37,800	28,000	22,400	15,060	12,320
5	3,600	.....	.....	48,600	36,000	28,800	20,520	15,840
6	6,000	.....	.....	.....	60,000	48,000	34,200	26,400
7	9,000	.....	.....	.....	.....	72,000	51,300	39,600
8	13,000	.....	.....	.....	.....	.....	74,100	57,200
9	18,000	.....	.....	.....	.....	.....	.....	79,200
10	23,000	.....	.....	.....	.....	.....	.....	.....
12	37,000	.....	.....	.....	.....	.....	.....	.....
14	55,000	.....	.....	.....	.....	.....	.....	.....
16	78,000	.....	.....	....	.....	.....	.....	.....

Column "A," steam rating in standard direct radiation for piping within buildings for all classes of systems.

Column " B," rating for wet return main of a gravity system.

Column " C," rating for main return of the Vacuum System. Thermostatic return values on radiators.

Column " D," rating for the main return when it is above the water line in a dry return gravity system.

Column " E," rating for the branch returns of the Vacuum System, and for the branch returns that are above the water line in a wet return gravity system.

Column " F," rating for the branch returns of a dry return gravity system.

Columns " G " and " H," ratings for gravity systems where the returns have unusual condensing capacity and for the returns in vacuum systems where jet water is used.

In one-pipe gravity work the *U. S. Treasury Dept.* uses a "wet" return much larger than that theoretically necessary in order to provide for the accumulation of core sand, scale, etc., with which such a return is eventually more or less obstructed. See Table 19. In these systems the return main is usually short as the steam main practically makes a circuit of the basement, dropping to the return near the boiler so that the extra cost of the larger pipe is not material. In such cases the return may be made equal to ½ the diameter of the steam main plus one-pipe size, so that a 3½" one-pipe steam main would have a 2" "wet" return instead of a 1" return main back to the boiler.

False Water Line. Returns that must run above the boiler water line, and therefore would naturally be "dry," may be changed to "wet returns," if necessary, by the use of a *false water line,* which is established at the end of the return just before it drops to enter the boiler. This inverted seal or dam should be protected against the possibility of freezing, and should be provided with a by-pass valve, as shown in Fig. 8–a. It is usually possible to locate the seal in the boiler room, near the boiler. It will be noted that the new water line on the branch return is now at a distance "0" above the actual boiler water line, which is still the water line of all other returns except this one.

Returns in Trenches. Return piping should always be run in as visible and open a manner as possible and never be buried below the basement floor. If it must be placed below the floor level, suitable trenches with removable covers should be provided (See Fig. 60 and detail) and provision made for draining the trench as well as the return main in the trench. In general, the trench should be arranged to grade back to the boiler pit and should discharge into same if possible.

All steam and return piping must be so designed and graded as to *drain freely when the blow off or drain connection at boiler is opened.* Any traps, seals or pockets must be avoided if possible, but if used by necessity they must be *separately drained.* Returns which drop below the floor for a short distance, at a doorway, for example, and then are relayed are liable to prove a serious source of trouble, due to leaks, sediment and freezing.

Radiation Required for Direct Heating. The method of proportioning the direct radiating surfaces or radiators for heating the various rooms of any building must be based upon the actual heat loss in B.t.u. from each room as well as on the rate at which the radiating surface can supply this loss. This heat loss is readily found by the methods explained in the chapter on " Heat Transmission of Building Materials and Direct Radiators," and the efficiency of various types of radiation is given in the same chapter.

If H, the heat loss from any given room in B.t.u. per hour is found, and Rs is the size of radiator in sq. ft. to just offset this loss then, for steam, the relation is $H = R_s \times K_s\ (t_s - t_a)$ or $R_s = H/[K_s\ (t_s - t_a)]$ where K_s is the coefficient of heat transmission for this radiator, t_s = temperature of steam, at 2 lb. pressure (usually), and t_a = temperature of air in room, usually 70° F.

Now H is made up of the heat transmitted through the walls, floors, ceilings and glass by conduction, and then dissipated, plus the in-leakage of cold air, which must be warmed to room temperature. $H = (W \times K_w + G \times K_g + V \times K_v)\ (t_a - t_o)$ where W = net area of outside wall, exclusive of openings in sq. ft., G = the "glass" area or openings for outside windows and doors, V = volume of air entering in cu. ft. per hour, K_w and K_g = coefficient of heat transmission for wall and glass respectively, while K_v = the B.t.u. to raise one cu. ft. of air through a temperature range of one degree; t_a and t_o = the temperature of air inside and outside of the building respectively.

DESIGN OF A ONE-PIPE GRAVITY STEAM HEATING SYSTEM

Heat Loss. The actual *calculations for the heat loss* from, and the square feet of radiation required for heating a factory office building (Figs. 59 and 60), where the outside temperature may go as low as −20°, are given in Table 25. The coefficients may be taken from tables of transmission

TABLE 25

HEATING DATA SHEET FOR FACTORY OFFICE BUILDING

HEAT LOSSES AND RADIATION

Heat Loss Factors B.t.u.)
Brick Wall, 13″ plain = 0.29 × 80 = 23.2 per sq. ft.
*1¾″ Wood Floor and ½″ Plaster Ceiling, { = 0.31 × 30 = 9.3 per sq. ft.; = 0.60 × 40 = 24.0 per sq. ft. }
Single Glass, ⅛″ thick = 1.07 × 80 = 85.6 per sq. ft.
Temperatures—lowest = −20°F.
Range = 70−(−10) = 80°, and all rooms figured on this basis and then modified by column 2. See column 14.
Radiation Factor = 1.70 (220−70) = 255 B.t.u.
Infiltration, 1/16″ Crack = 2.4 × 80 = 192.0 per lineal ft.
Windows double hung—Door 4′ × 9′

Floor	Room—Designation	Temp. F.°	Net Volume, Cu. Ft.	Wall Area (Net) Sq. Ft.	Floor or Ceiling Sq. Ft.	Glass Area (Opening) Sq. Ft.	Transmission Loss—B.t.u.: Wall Loss 23.2 × Col. 4	Floor or Ceiling loss 9.3 / 24.0 × Col.5	Glass Loss 85.6 × Col. 6	Lin. Ft. of Crack, Worst Side	Infiltration Loss 192 × Col. 10	Total Loss, B.t.u.	Steam Rad. Sq. F. Col. 12/255	Col. 13 Cor. for Temp. in Col. 2	No. of Rads	Height Rads	Cols. Rads	Area of 1 Rad	Total Rad. Sq. Ft. Actual Surface
0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15				16
1	Sample.	80°	10,080	852	864	180	19,850	8,030	15,400	†81	15,540	59,820	234	263	4	32″	2	66⅔	267
1	Hall.	60°	2,595	99	216	45	2,300	2,010	3,850	26	5,000	15,660	61	53	1	38	3	. . . .	60
1	Labor.	60°	4,320	378	360	90	8,780	3,340	7,700	54	10,360	30,180	118	118	2	32	2	60	120
1	Store.	50°	2,520	288	210	60	6,700	1,950	5,130	27	5,180	18,960	75	56	1	32	2	. . . .	53½
1	Toilet.	50°	900	90	75	30	2,090	700	2,560	27	5,180	10,530	41	31	1	32	2	. . . .	30
2	Mgr's.	70°	4,320	393	360	75	9,130	8,650	6,400	50	9,600	33 780	132	132	2	32	2	63⅓	127
2	Hall.	60°	2,590	119	216	25	2,760	5,200	2.140	25	4,800	14 900	58	51	1	32	2	. . . .	40
2	Gen. Off.	70°	10,080	882	864	150	20,250	20,700	12,800	†75	14,400	68,150	267	267	4	32	2	66⅔	267
2	Supts.	70°	4,320	393	360	75	9,130	8.650	6,400	50	9,600	33,780	132	132	2	32	2	66⅔	133
	Totals.		41,725									285,760							1097½

NOTES.—*It is assumed basement and attic are at 40° and 30° respectively in coldest weather.
† Use not less than 1½ total crack for 3 or more sides exposed
Radiation to be supplied by Boiler = 1100 + 25% for pipe line losses = 1375 square feet.

coefficients for building materials, or may be calculated as already indicated under heat loss from buildings in the same chapter. Likewise the coefficients for radiation may be taken from tables of transmission coefficients for direct steam radiators, or may be calculated from test data as indicated in this example.

It will be noted that the temperature range is based on −10° F. outside, which is 10° above the lowest on record. The attic space is assumed at a mean between the inside and outside temperatures, and the basement likewise, and allowance made for loss through first floor to basement and second floor ceiling to attic.

By reference to Table 25, which is practically self-explanatory, it will be seen that the volume, net outside wall, and total "glass" areas have been computed for each room, as well as floor and ceiling, areas where heat loss through same is to be allowed for. See Columns 3, 4, and 5. The

transmission losses are then found by multiplying by the proper heat loss factors, Columns 7, 8, and 9, which were determined as follows:

(1) For outside brick walls 13″ thick and unplastered,

$$K = \frac{1}{\frac{1}{1.4} + \frac{1}{4.2} + \frac{13}{5}} = 0.29 \text{ B.t.u}$$

(2) For single glass 1/8″ thick

$$K = \frac{1}{\frac{1}{1.5} + \frac{1}{4.5} + \frac{.125}{3}} = 1.07 \text{ B.t.u.}$$

(3) For wood floor $1\frac{3}{4}$″ thick (no ceiling below).

$$K = \frac{1}{\frac{2}{1.4} + \frac{1.75}{1.0}} = 0\ 31$$

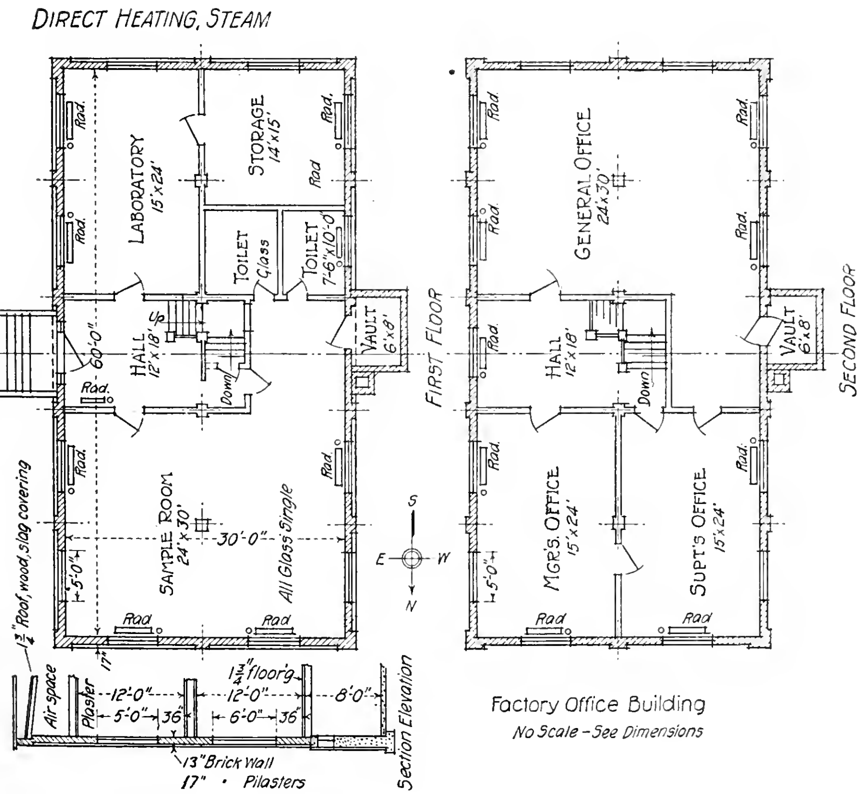


FIG. 59.

(4) For lath and plaster ceiling only (no attic floor).

$$K = \frac{1}{\frac{1}{0.93} + \frac{1}{1.86} + \frac{0.5}{8.0}} = 0.60$$

NOTE.—$K_2 = 2K_1$, or $0.93 \times 2 = 1.86$ for upper side of ceiling.

The infiltration losses are based on a 1/16″ crack as found on most exposed side of room, but, for rooms with more than two sides exposed, use at least ½ of total crack, or, $2.4 \times 80 = 192$ B.t.u. per ft. per hr.

The radiation coefficient by table, see chapter on "Heat Transmission of Radiators" for 2 column, 32″ high radiators is 1.70 B.t.u., and the radiation factor is, therefore, 1.70 (220 − 70) = 255.0 B.t.u.

If the transmission coefficients have not been determined under *moving* air conditions then an exposure factor must be used for increasing the wall and glass transmission losses.

Exposure factors may be based on the following: N. = 1.32, E. = 1.12, S. = 1.00, W. = 1.20, N.E. = 1.22, N.W. = 1.26, S.E. = 1.06, S.W. = 1.10, all sides = 1.16.

The infiltration loss given in Column 11 is based on the lineal feet of crack at each window as already explained for heat loss from buildings. This loss has been found to average 2.4 B.t.u. per lineal ft. of 1/16″ crack, for a temperature difference of one degree with average wind velocity. Area of glass in square feet may be taken equal to the lineal feet of crack around same for a double hung window of average proportions.

Radiation Required. The total B.t.u. loss per hour is given in Column 12 for each room as indicated, and in column 13 is given the sq. ft. of radiation having a coefficient of transmission K as indicated in the table of coefficients in the chapter on "Heat Transmission of Radiators."

It will be found that sizes of commercial radiators will not in general coincide with the values for radiating surface found in column 13, and, moreover, in many cases it will be necessary, and desirable, to subdivide these values for radiation into smaller units, placing several radiators in one room, especially if large, to secure better control and distribution of heat. Hence, by referring to tables which give the commercial sizes for what is known as the Peerless cast-iron radiator, and using the 2 column 32″ high type, we may select the actual radiation listed in column 15. This is laid out to scale on the 1st and 2d floor plans, and indicated with main and riser connections on the basement heating plan. (Figs. 59 and 60.)

Boiler Size. The boiler size required will depend, as already indicated under the chapter on "Heating Boilers," upon the total amount of radiation installed, plus the allowance to be made for the steam and return mains, risers, fittings, etc., as equivalent radiating surface. If the mains are covered this equivalent is taken as 25 per cent of the direct radiation installed, hence we will require a boiler with a rating of at least 1375 sq. ft. of direct steam radiation. Radiation loss from mains may be actually figured if desired, using square feet of surface and coefficients of radiation for pipe as covered.

If the boiler burns a fair grade of coal developing 12,000 B.t.u. per lb. as its heat value, and has an efficiency of 60% when burning 6.7 lb. per sq. ft. of grate per hour, we must have a grate area, $G = \frac{1375 \times 254}{12000 \times 0.60 \times 6.7} = 7.24$ sq. ft. By reference to the table of cast-iron boiler capacities in the chapter on "Heating Boilers," it will be seen that an *S–25–7 Ideal* cast-iron boiler having a grate area of 8.16 sq. ft. is required.

The *depth of fire pot* of this boiler should also be checked as indicated under the chapter on "Heating Boilers," in order to ascertain whether it will hold the necessary 8 hour firing charge plus 20% for rekindling.

Chimney Size. The chimney size is next ascertained from curves of chimney capacities

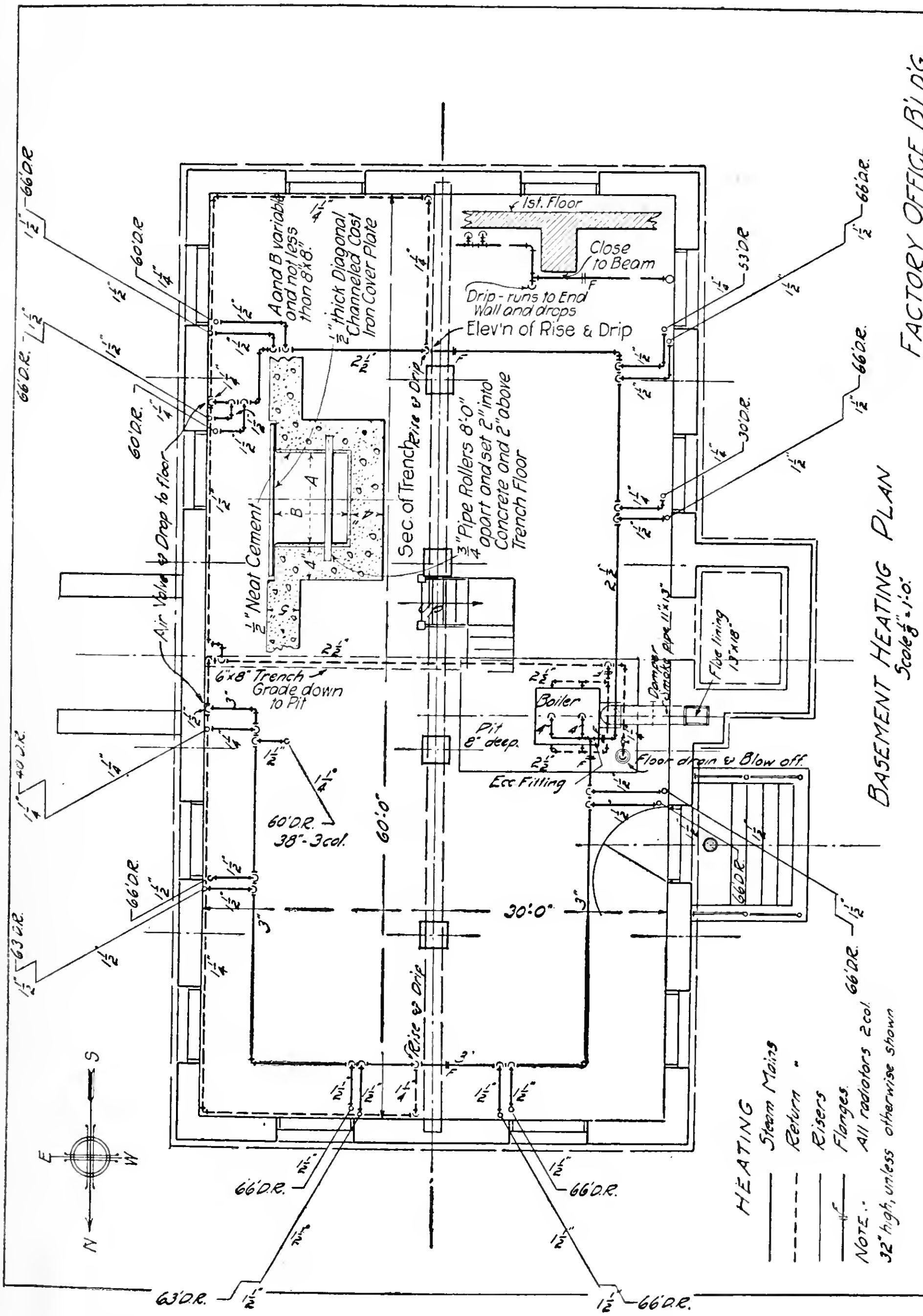

FACTORY OFFICE B'LD'G.
BASEMENT HEATING PLAN
Scale 1/8"=1'-0"
HEATING
Steam Mains
Return "
Risers
Flanges
NOTE: All radiators 2 col. 32" high, unless otherwise shown
1st Floor
Close to Beam
Drip - runs to End Wall and drops
Elev'n of Rise & Drip
A and B variable and not less than 8"x8"
1/2" thick Diagonal Channeled Cast Iron Cover Plate
1/2" Neat Cement
Sec. of Trench
3/4" Pipe Rollers 8'-0" apart and set 2" into Concrete and 2" above Trench Floor
Air Valve & Drop to floor
6"x8" Trench Grade down to Pit
Boiler
Pit 8" deep.
Ecc Fitting
Floor drain & Blow off.
Damper
Smoke pipe 11"x13"
Flue lining 13"x18"
60'-0"
30'-0"
60' D.R. 38"-3col.
Rise & Drip
S
E
W
N

FIG. 60.

in the chapter on "Chimneys for Heating Boilers," for a stack of the height shown in the elevation section, and a flue lining selected from table of flue linings. The height of chimney is measured from the grate level to top of masonry which should be at least 2′–0″ above the ridge, or highest point of the roof, in this case 44′–0′, and hence by curves a flue 13″ sq. or 169 sq. in. in area is required, so that a 13″ x 18″ fire clay flue lining is specified. Net area = 10¾″ x 15¾″ or 170 sq. in.

Heating Plan. The main piping, branches, and risers are laid out as indicated in Fig. 60, the sizes being taken from Tables 19 and 21, and all first floor risers are run separately, as shown, which practice is strongly recommended in one pipe systems. It will be noted that by laying the risers down on the basement plan it is possible to show the size on this one sheet of every piece of pipe, valve and fitting for the entire installation.

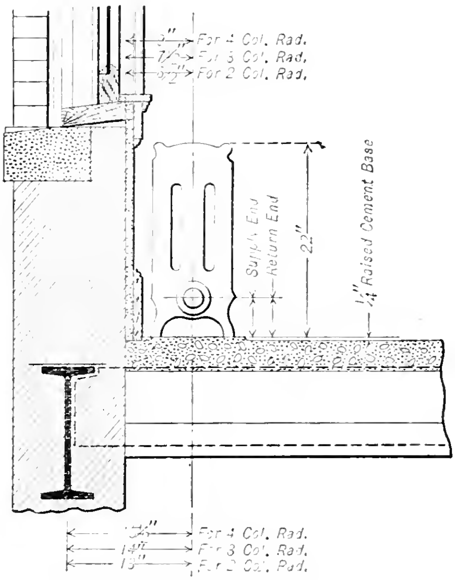


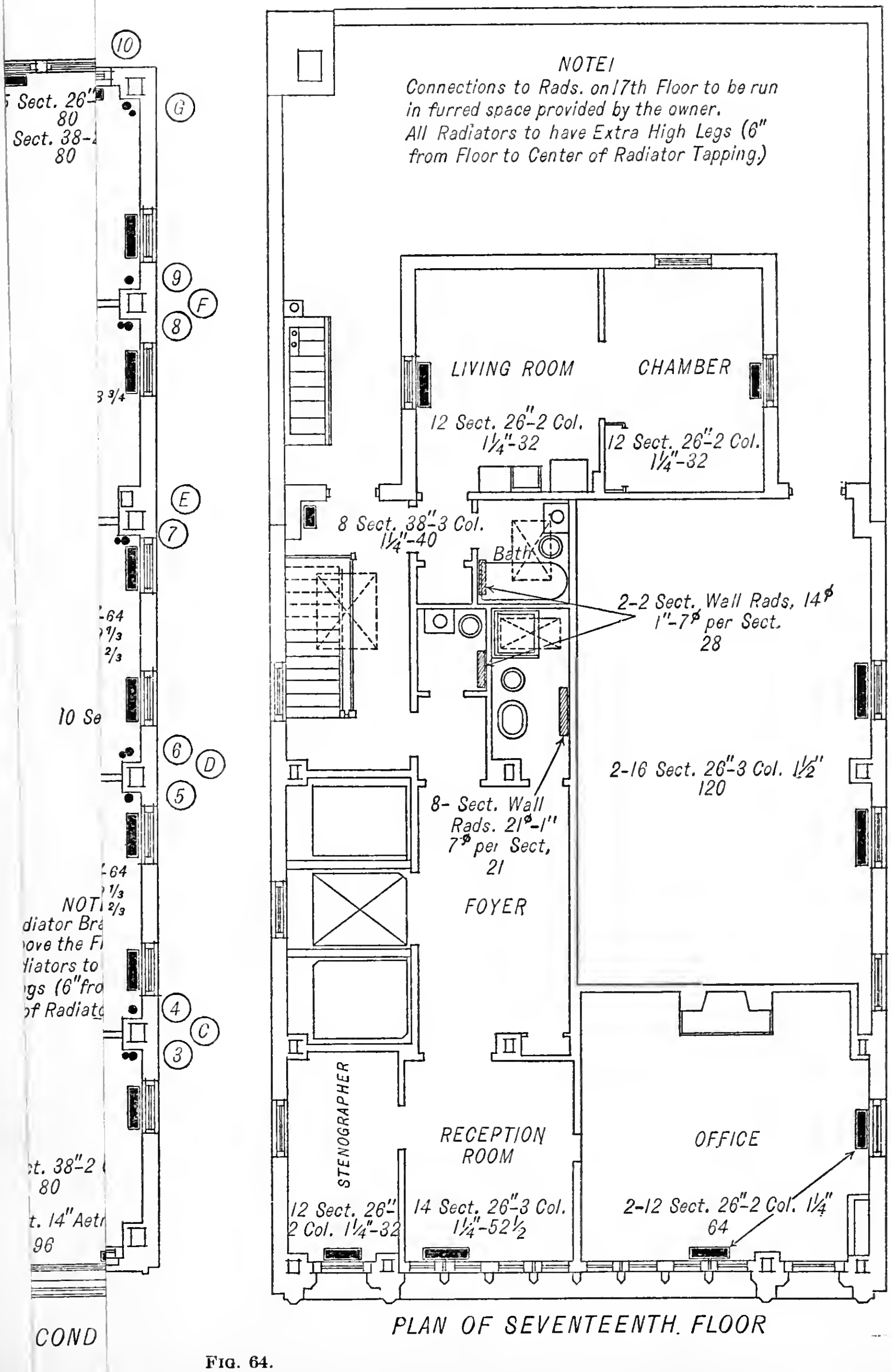
FIG. 61. "ROUGHING IN" PLAN FOR RADIATORS.

The boiler is shown in a shallow *pit* 8″ deep and the 2½″ return has been brought back in a *trench* with a cast iron cover plate as shown by detail section of same on the heating plan. This was done in order to avoid crossing above the floor, and at the same time not obstruct doorways. The main has been "relayed" at two points where it passes under beam. See "Elevation of Rise and Drip." (Fig. 60.) If the basement space is not valuable and the end of the main is at least 18″ above boiler water line these relays are not necessary. These drips have been run on basement walls below windows and connected to return mains at the drops. Automatic air valves are indicated at each drop. (See also Fig. 2.)

The breeching or smoke connection and flue lining are all indicated on this plan and sizes given.

The radiators are shown always on the floor plans (Fig. 59), and should be drawn to scale, and the location of valves on same are indicated and must conform to riser locations as shown on heating plan.

A key to the heating symbols used is shown in the corner of basement plan and complete data given as to height and number of columns for all radiators. See the chapter on "Prepara-



NOTE!
Connections to Rads. on 17th Floor to be run in furred space provided by the owner.
All Radiators to have Extra High Legs (6" from Floor to Center of Radiator Tapping.)
LIVING ROOM
CHAMBER
12 Sect. 26"-2 Col. 1¼"-32
12 Sect. 26"-2 Col. 1¼"-32
8 Sect. 38"-3 Col. 1¼"-40
Bath
2-2 Sect. Wall Rads. 14ϕ 1"-7ϕ per Sect. 28
2-16 Sect. 26"-3 Col. 1½" 120
8- Sect. Wall Rads. 21ϕ-1" 7ϕ per Sect, 21
FOYER
STENOGRAPHER
RECEPTION ROOM
OFFICE
12 Sect. 26"-2 Col. 1¼"-32
14 Sect. 26"-3 Col. 1¼"-52½
2-12 Sect. 26"-2 Col. 1¼" 64
PLAN OF SEVENTEENTH FLOOR

Fig. 64.

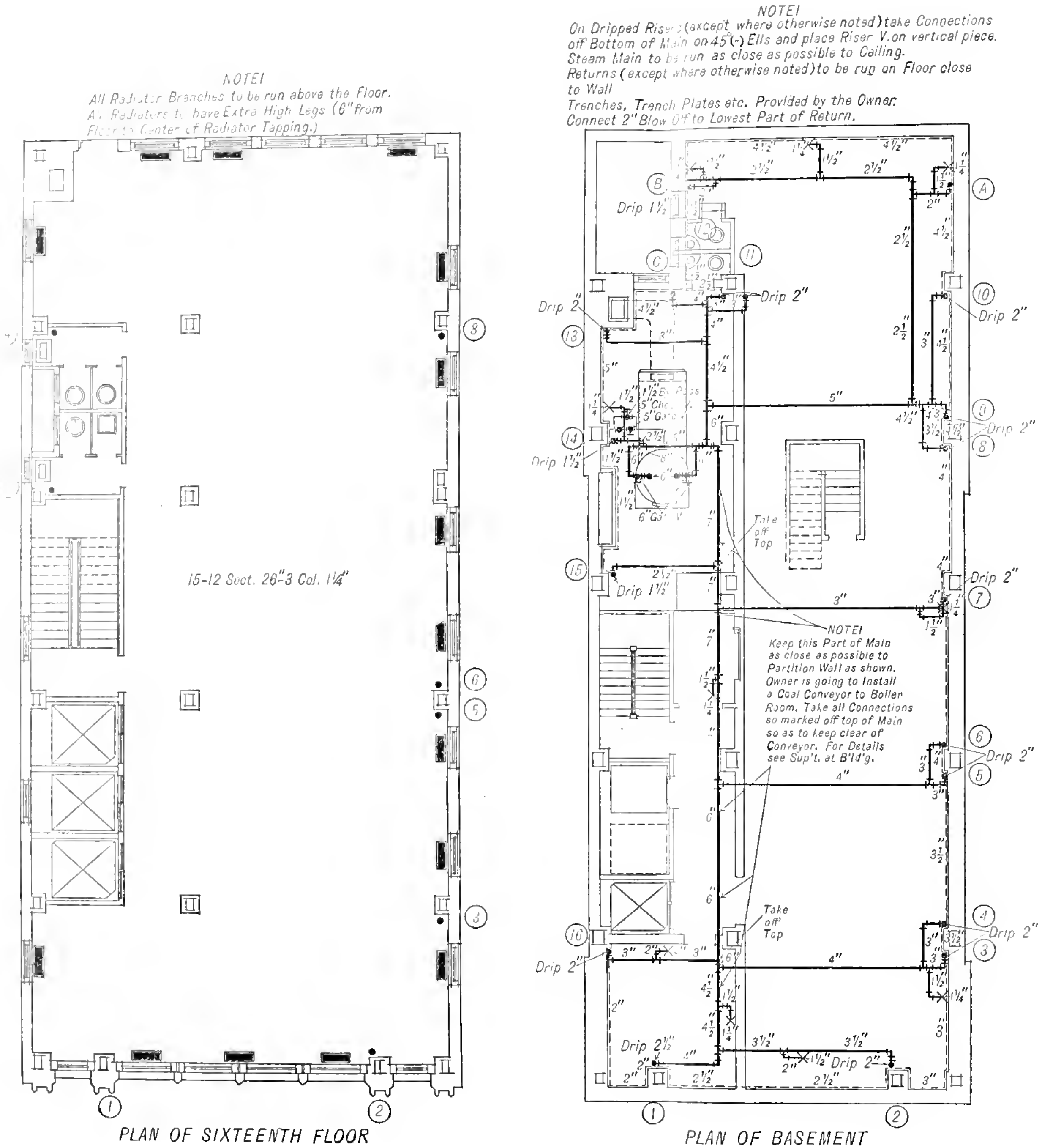
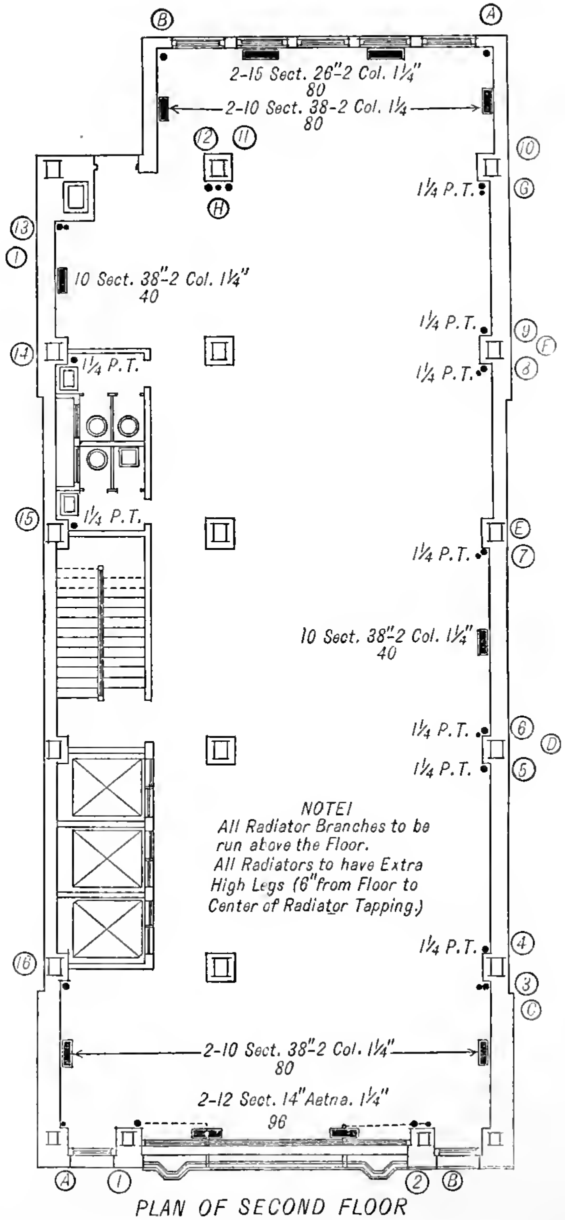
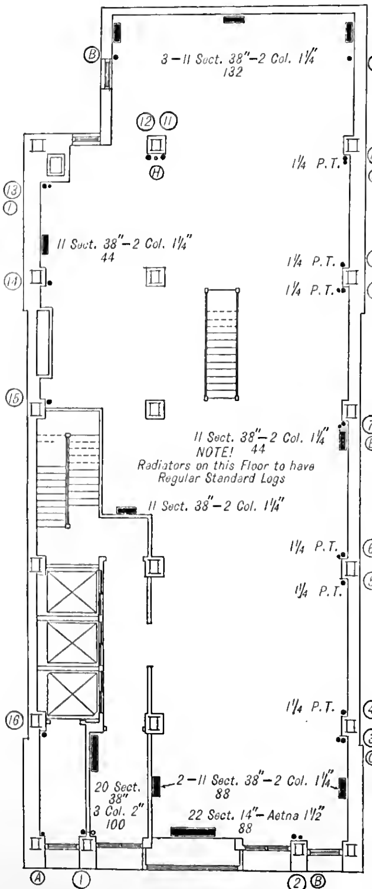

NOTE!
All Radiator Branches to be run above the Floor.
All Radiators to have Extra High Legs (6" from Floor to Center of Radiator Tapping.)
15-12 Sect. 26"-3 Col. 1¼"
PLAN OF SIXTEENTH FLOOR
NOTE!
On Dripped Risers (except where otherwise noted) take Connections off Bottom of Main on 45° (-) Ells and place Riser V. on vertical piece.
Steam Main to be run as close as possible to Ceiling.
Returns (except where otherwise noted) to be run on Floor close to Wall
Trenches, Trench Plates etc. Provided by the Owner.
Connect 2" Blow Off to Lowest Part of Return.
Take off Top
NOTE!
Keep this Part of Main as close as possible to Partition Wall as shown. Owner is going to Install a Coal Conveyor to Boiler Room. Take all Connections so marked off top of Main so as to keep clear of Conveyor. For Details see Sup't. at B'ld'g.
Take off Top
PLAN OF BASEMENT

FIG. 62.

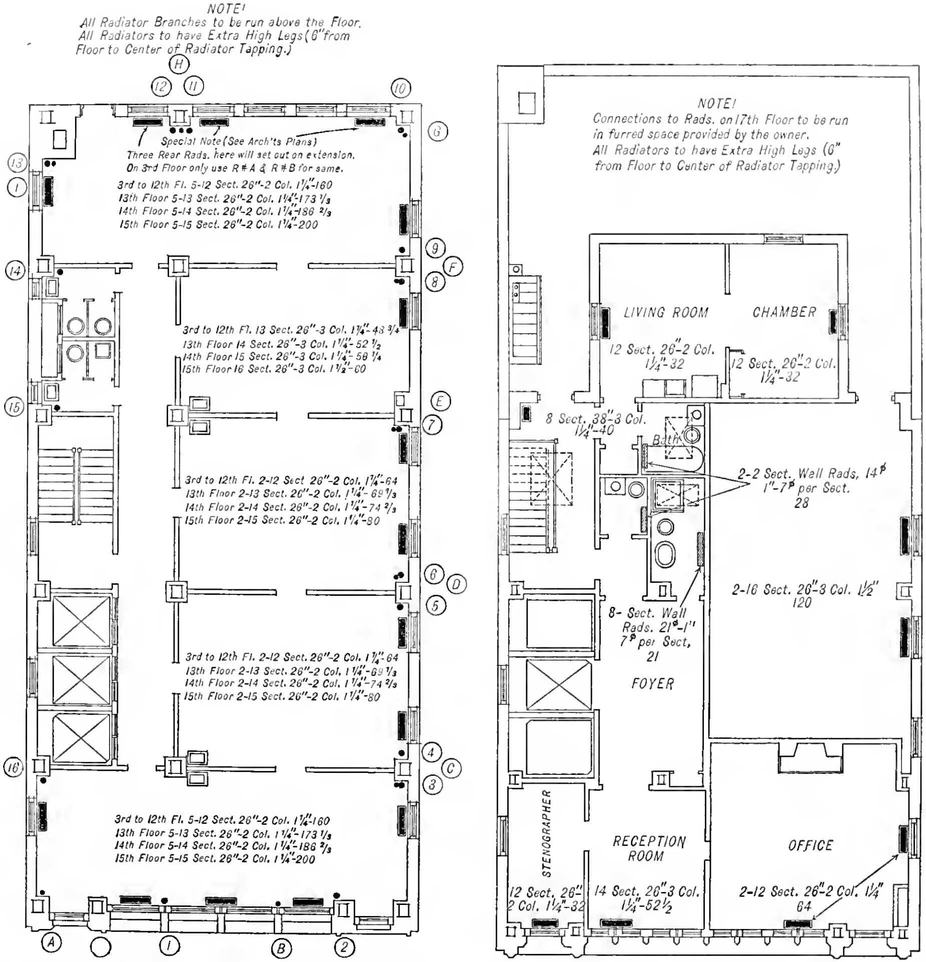

2-15 Sect. 26"-2 Col. 1¼"
80
2-10 Sect. 38-2 Col. 1¼
80
10 Sect. 38"-2 Col. 1¼"
40
10 Sect. 38"-2 Col. 1¼"
40
NOTE!
All Radiator Branches to be run above the Floor.
All Radiators to have Extra High Legs (6" from Floor to Center of Radiator Tapping.)
2-10 Sect. 38"-2 Col. 1¼"
80
2-12 Sect. 14" Aetna. 1¼"
96
PLAN OF SECOND FLOOR

FIG



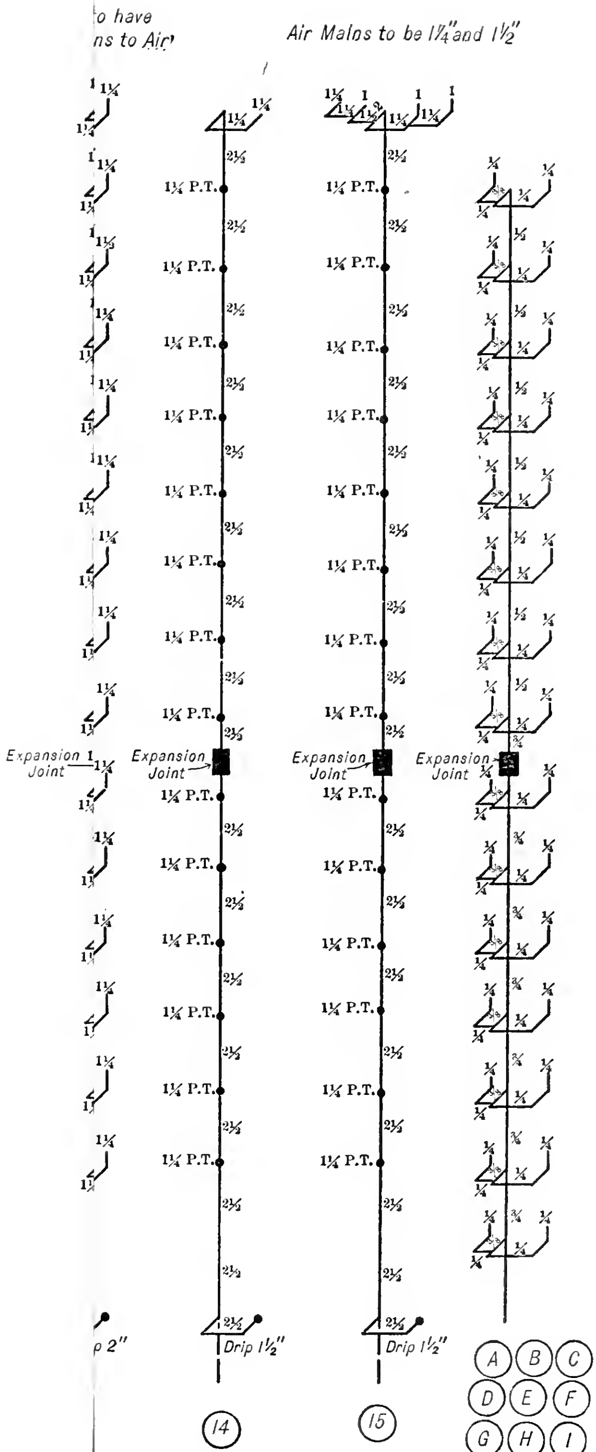
PLAN OF FIRST FLOOR

FIG. 63.



PLAN OF SEVENTEENTH FLOOR

FIG. 64.

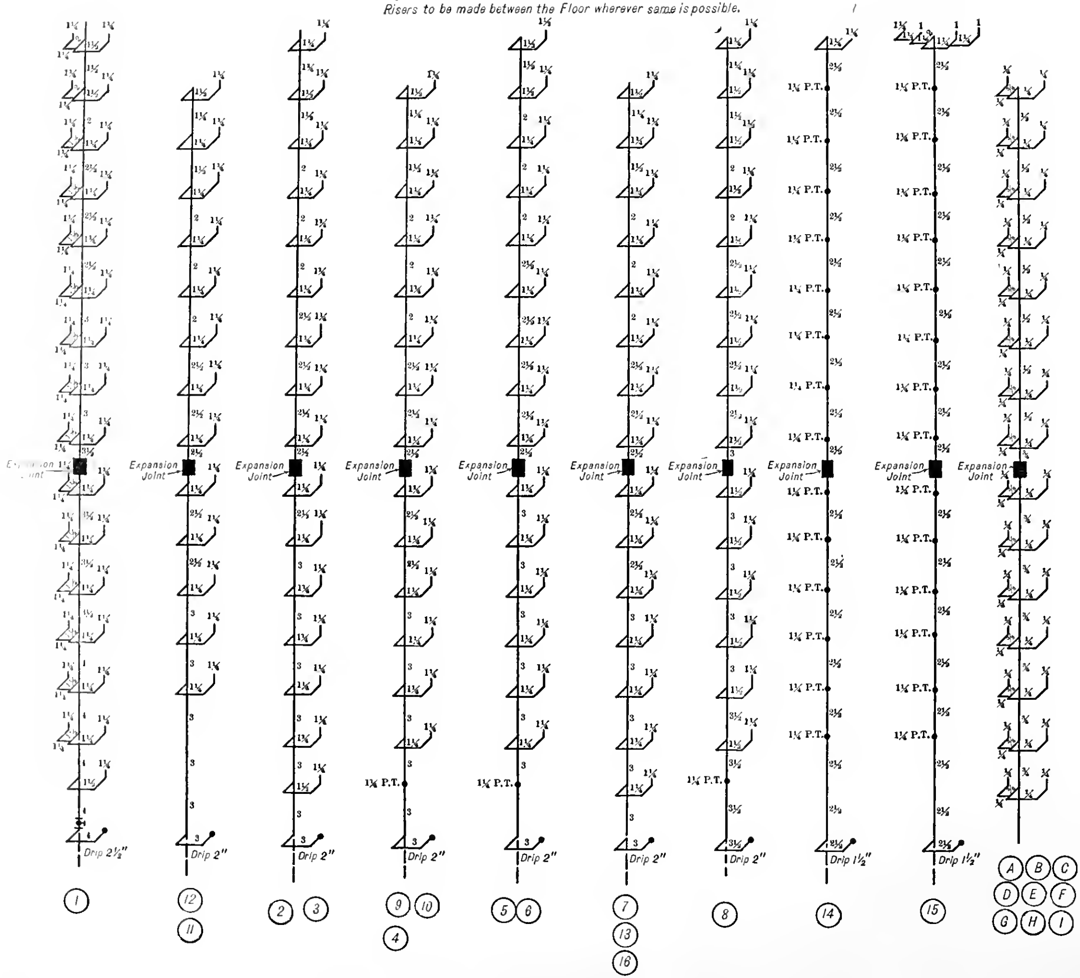


o have
ns to Air
Air Mains to be 1¼" and 1½"
Expansion Joint
Expansion Joint
Expansion Joint
Expansion Joint
1¼ P.T.
2½
Drip 1½"
Drip 1½"
p 2"
14
15
A B C
D E F
G H I

NOTE

All Radiator Branches to be run above the Floor. All Rads. to have High Legs 6" from Floor to Center of Rad. Tapping. Connections to Air Risers to be made between the Floor wherever same is possible.

Air Mains to be 1¼" and 1½"



RISER SHEET
No Scale

tion of Plans and Specifications" for approved Heating and Ventilating Symbols as used by the *U. S. War and Treasury Departments.*

A practical commercial specification for this installation is also given in the chapter on "Preparation of Plans and Specifications." Also see Appendix.

Direct Heating System for High Building. A typical installation of a one-pipe low pressure gravity circulating direct steam-heating system in a modern Loft Building of seventeen stories is shown in Figs. 62 to 65 inclusive. This building is located at Nos. 56 and 58 West 45th St., New York City, and is typical of this class of buildings now so common in all large cities.

All radiators are served with one-pipe up-feed risers, and above the 12th floor the size of the radiators is increased, as shown, to provide for greater wind movement and increased heat loss.

The *Location* of radiators so that they will be properly spaced from the finished wall can be determined from the "roughing in" plan for radiators shown in Fig. 61.

A positive air-removal system is connected to each radiator, with an electrically driven air-line pump or exhauster in the basement.

The sizes of all steam and air risers are shown on the riser sheet (Fig. 65), and the designating numbers given in the small circles indicate the corresponding locations on the various floor plans.

CHAPTER X

DIRECT HOT-WATER HEATING

Systems in Use. Systems for heating with direct hot water radiators, like the direct steam heating systems, may be divided into two general classes, the first of which includes all those systems *operating by gravity only*, depending on the difference in density of the water columns in the flow and return lines to produce circulation. The second class includes those systems in which a *forced circulation* is maintained by means of a pump placed on the return line just before it enters the boiler or heater. These latter systems are employed usually only in large installations or in district heating service and will be considered in the chapter on " District Heating."

GRAVITY HOT-WATER HEATING SYSTEMS

The gravity systems are divided into the *up-feed systems*, using basement mains, and the *down-feed systems*, using overhead or attic mains. The up-feed systems may have either a one-pipe basement main or two-pipe basement mains, and the latter type may have either a *direct* or a *reversed* return main. (See Figs. 1 and 2 for reversed return.) The down-feed systems may have either *single or double risers*, as shown in Figs. 6 and 7. Either system may be operated with an *open* or *closed expansion tank*, as shown in Figs. 10 to 14.

In general, the down-feed or overhead systems are more positive, permit the use of smaller mains and riser, and provide for the automatic removal of air from the radiators and piping. It is necessary, however, that the head-room or clear space in the attic should be at least 4 or 5 feet if the overhead mains and branches are to be properly installed. It is sometimes possible to run the overhead mains at the ceiling of the top floor, and in such cases the above restriction does not apply.

The under-feed systems are used where basement space is available, and of little or no value, and the radiation is located on two or more floors, or where attic space is so limited that it would not be possible to install overhead mains and branches. Under-feed systems are liable to prove unsatisfactory in buildings less than two stories in height, as the motive head with radiators on the first floor only is so slight that faulty or deficient circulation is quite likely to result.

The Up-Feed One-Pipe System. The up-feed one-pipe system is in very general use today, and is employed almost exclusively by the *U. S. Treasury and War Depts.* whenever up-feed hot-water systems are to be installed. In this system, as shown in Figs. 11 and 13, the supply main rises close to the basement ceiling just above the boiler, and grades down in the direction of flow, with a uniform grade of 3/4″ in 10 feet. Branches are taken from the top of this main for supplying flow risers and the return branches are made into the side or bottom. (Fig. 3.) Flow connections should always be made from the top or at an angle of 45° in the case of branches near the boiler, or for branches supplying only upper floor radiators.

It will be seen that in the case of branches supplying radiators on all floors the upper floor radiators may be made to "pull" or augment the circulation of the first floor radiators by taking the basement branch for the former from the side of the branch running to the latter radiator. The first floor branch is usually run full size all the way to favor the lowest radiator, as shown in Fig. 13. After having served all the radiator branches the main drops and returns to the boiler, continuing the same size for the entire circuit. Connections (Fig. 11) to radiators should be made at the *top* on the supply end, using a union elbow, and at the *bottom* on the return end,

HOT WATER HEATING – PIPING SIZES – OPEN TANK
(UP FEED WITH BASEMENT MAINS)

TABLE 1

MAINS Up to 100'-0"		
Pipe Size	Direct Radiation #	Indirect Radiation #
$1\frac{1}{4}$	135	100
$1\frac{1}{2}$	220	135
2	350	225
$2\frac{1}{2}$	460	320
3	675	500
$3\frac{1}{2}$	850	650
4	1100	850
$4\frac{1}{2}$	1350	1050
5	1700	1350
6	3600	2900
7	4800	3900
8	6200	5000
9	7700	6300
10	9800	7900
12	14000	11400

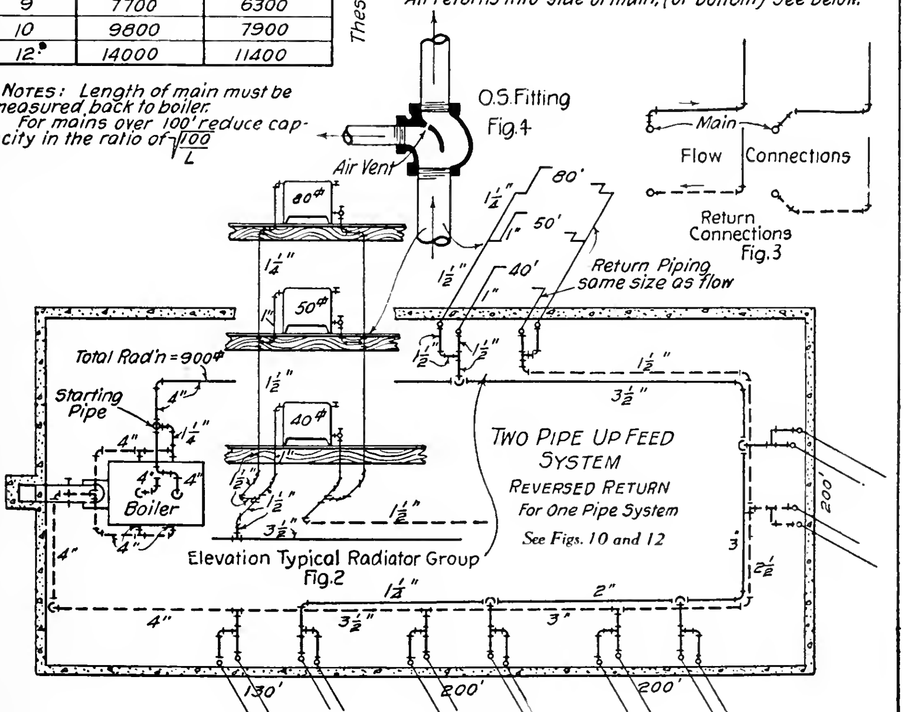
These Tables by J.J.Hogan, to be used for either one or two pipe work.

TABLE 2

BRANCHES & RISERS				
Pipe Size	Floor First	Second	Third	Fourth
	Direct Radiation #			
$\frac{3}{4}$	30	45	55	70
1	60	75	85	95
$1\frac{1}{4}$	110	120	135	150
$1\frac{1}{2}$	180	195	210	230
2	290	320	350	370
$2\frac{1}{2}$	400	490	525	550
3	620	650	690	730
$3\frac{1}{2}$	820	870	920	970
4	1050	1120	1185	1250
$4\frac{1}{2}$	1325	1400	1485	1560

NOTES: First floor supply riser connections to be taken from top of main, and risers above first at 45°. All returns into side of main, (or bottom) See below.

NOTES: Length of main must be measured back to boiler. For mains over 100' reduce capacity in the ratio of $\sqrt{\frac{100}{L}}$



Elevation Typical Radiator Group Fig. 2

Basement Heating Plan. Fig. 1

using a quick opening hot water radiator valve with union connection. By this arrangement only one valve is required to control the radiator.

Since the temperature of the water in the one-pipe main gradually drops, due to the return of water at a lower temperature from the radiators served in the course of the main around the building, it is advisable to increase the last radiators on the main from 5 to 10% in area and increase the size of branch and riser connections at the end of the main by one pipe size.

Pipe sizes may be taken from Tables 1 and 2 or from the Friction Pressure Loss Chart (Fig. 19), which is given under the theoretical discussion of pipe sizes. In using the tables all mains must be measured back to the boiler, and risers to any floor are proportioned to supply all the radiation above that floor as well as the radiator actually installed on this floor, as shown in Figs. 11 and 13.

The Up-Feed Two-Pipe System. The up-feed two-pipe system is also in very general use, and if installed with a "reversed" return, as shown in Figs. 1 and 2, will give good results. If a "direct" return is used so that the water circulates first through the radiators nearest the boiler, and then through each succeeding group in turn, the ends of mains will be slow in warming up, the last radiators may be cold, and the system prove unsatisfactory.

With the "reversed" return system each group of radiators has exactly the same length of water travel, and hence the resistance to be overcome is practically the same irrespective of the distance of the radiator group from the boiler. It will be noted that the return begins at the first radiator served and flows in the *same* direction as the flow main, increasing in size while the latter decreases. The flow main grades *up* uniformly 3/4″ in 10′–0″, and the return grades *down* toward the boiler with the same pitch.

Pipe sizes may be taken from Tables 1 and 2 as in one pipe work, and the main size reduced or increased as rapidly as the change in radiation supplied will permit. See also the theoretical discussion of pipe sizes, and the chart, Fig. 19.

It is also customary in government work to install a *starting pipe* (Fig. 1), between the main flow and return at the boiler, in under-feed systems. This pipe ranges from 1¼″ to 2½″ in size depending on the capacity of the boiler, and is intended to assist in the establishment of an initial circulation between flow and return headers, even before the water in the mains is moving.

The Down-Feed System. The principal details of a typical down-feed system are shown in Figs. 5 to 8, and it will be noted that after leaving the boiler the "flow" main is carried overhead to the attic as soon as possible through a single "main riser," the size of which can usually be made much smaller than that of the horizontal mains in basement or attic to which it connects. (See Tables 3 and 4.) The main riser must be firmly supported (Fig. 8), and is provided with an enlarged chamber to permit of the rapid separation of air or other gases which tend to collect at the top of same, and are removed through the vent line to the expansion tank.

The flow main in basement grades *up* from the boiler to foot of main riser, and in the attic it grades *down* from the point of entrance at head of main riser with a uniform grade of 3/4″ in 10′–0″ in both cases.

The overhead branches to drop risers are taken from the side or bottom of the overhead main, and by use of double elbows, as shown, can be properly graded and also arranged for expansion. If *single risers* are to be run, connections are made as shown in Fig. 6 and sizes may be taken from Table 5, care being taken to proportion each portion of the riser for the amount of radiating surface it must serve both as a supply and a return line. If *double risers* are to be run, as shown in Fig. 7, they may be made somewhat smaller than single risers, and need only be proportioned to act as flow or return risers, respectively. (See Table 5.) In the overhead systems the pipe size for the same size of radiator on any floor is the same, as given in Table 5. By reference to the figures it will be seen that with either single or double drop risers there is always a circulation of water through the flow riser whether any or all the radiators are shut off or not.

Special attention should be given to the way in which the radiator branches are connected into the risers, to readily provide for the escape of air up the riser and into the overhead main.

HOT WATER HEATING-PIPING SIZES-OPEN TANK
(DOWN FEED WITH ATTIC MAINS)

Main and Drop Riser Tables by C. B. Thompson

Table 3

MAIN RISER	
Pipe Size	Rad'n. sq. ft.
$1\frac{1}{4}$	200
$1\frac{1}{2}$	350
2	600
$2\frac{1}{2}$	1200
3	2000
$3\frac{1}{2}$	2500
4	3500
$4\frac{1}{2}$	4500
5	6000
6	10000
7	12000
8	15000

NOTE—Chamber at head of Riser made of a bushed tee having the run twice the diam. of riser, and outlet same size as attic main.

TABLE 4

MAINS 100' long	
Pipe Size	Rad'n. sq. ft.
$1\frac{1}{4}$	150
$1\frac{1}{2}$	250
2	350
$2\frac{1}{2}$	540
3	900
$3\frac{1}{2}$	1300
4	1800
$4\frac{1}{2}$	2400
5	3200
6	5000
7	7200
8	10000
10	13000
12	16000

TABLE 5

DROP RISERS		
Pipe Size	Single Riser	Double Riser
$\frac{3}{4}$	Direct 25	Rad'n ⌀ 50
1	80	120
$1\frac{1}{4}$	120	200
$1\frac{1}{2}$	200	350
2	400	600
$2\frac{1}{2}$	750	1200
3	1000	2000

NOTES—Radiator tappings same as pipe size, and for same size radiator pipe size is the same for any floor. Increase riser one size at end of overhead main.

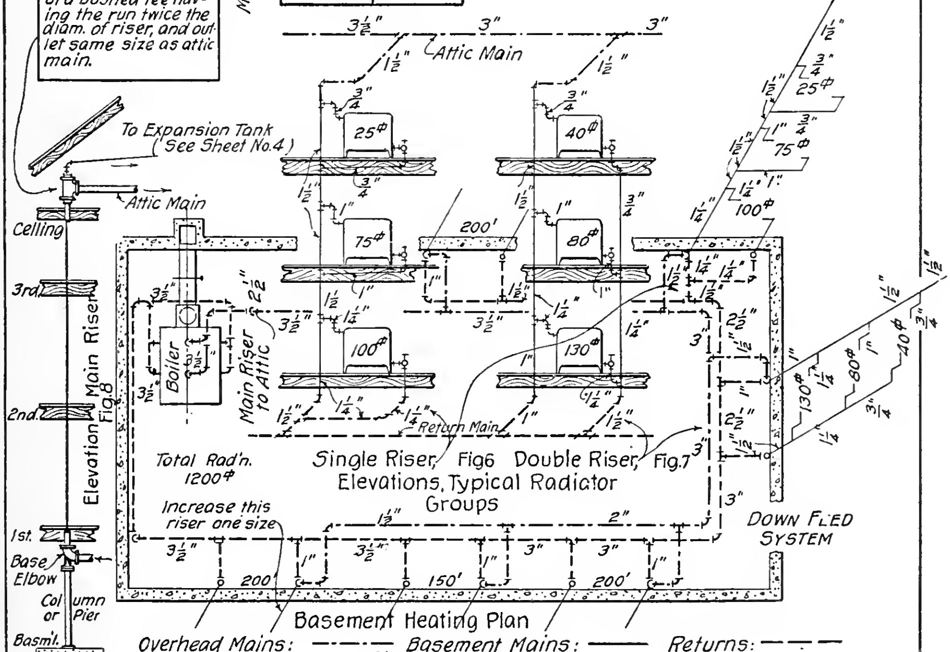


Fig. 5

No air valves are required with a down-feed system properly designed. Each radiator is connected on the flow side with a union elbow, and by a quick opening hot-water radiator valve with union on the return. If the connections are made as shown, only one radiator valve will be required to control the radiator.

The flow main size decreases as branches are taken off, and the return main in basement increases correspondingly. (See Table 4 or use Equalization Table 6.) The return main begins just under the first radiator served by the overhead main, and is run "reversed" so that its flow is in the same direction as that of the overhead main, thus giving equal length of water travel to any radiator in the building, and therefore equal resistance to flow, which is one of the chief advantages of this system.

Equalization Table. In Federal building work *N. S. Thompson* makes use of the following Equalization Table in proportioning mains and risers serving more than one radiator in both up-feed and down-feed systems.

The equalizing numbers represent the relative capacities of the different sizes of pipes for the same friction pressure loss per 100 feet of run, and are proportional to the 5/2 powers of the diameters. Thus the weight of water flowing varies as shown by the relation, $W = K\ d^{5/2}$, where W = weight, K = a constant, and d = pipe diameter.

TABLE 6

EQUALIZATION TABLE FOR MAINS AND RISERS

Ins.	Ins.	Ins.	Ins.	Ins.
½ = 2	¾ = 5	1 = 10	1¼ = 20	1½ = 30
2 = 60	2½ = 110	3 = 175	3½ = 260	4 = 380
5 = 650	6 = 1050	7 = 1600	8 = 2250	

Example. A 1¼", 1½" and 2" pipe have a value of 110 units, and hence are equivalent to the carrying capacity of a 2½" main.

Details of Piping Systems and Connections for Direct Hot-Water Heating. The distinctive piping details of each system of hot-water heating have been discussed under that system, as described in the preceding paragraphs.

In general all main piping and branches must be *uniformly graded*, as already indicated, and ample provision made for *expansion* and *contraction*, and the ready *removal of air* from all parts of the system. Air traps or pockets in a hot water system are fully as serious as water pockets in a steam system. Hence a hot water main grading down in the direction of flow cannot be relayed unless an air outlet is provided at the top of the relay. If the main is reduced in size at any point an *eccentric fitting* must be used to keep the *top* of the large and small main in the same plane and avoid an air pocket. Not only must all the piping be designed to permit the removal of air, but *free and complete drainage* of water must be provided for as well, so that when the drain or blow-off cock is opened at the boiler the entire system can be emptied of water.

If branch mains are taken from a *header* at the boiler they must all rise to the *same elevation* so that the tops of all the branches will lie in the same plane as they start away from the boiler.

The fittings on all main piping and branches must be of the long sweep pattern, and all pipe should be carefully reamed to remove burrs and sharp edges. Where the same riser supplies radiators on two or more floors the branches to the radiators on the intermediate floors may be connected with special tees (Fig. 4) known as O. S. fittings, with a deflector arranged to divert the current of flow into the outlet of the tee, and thus favor the radiators on the intermediate or lower floors.

The *branches* and *risers* have already been considered under each system as described. In no case should branches longer than 9" be allowed above the floor for connecting either steam or water radiators, and if a greater length is necessary the branch must run in the floor

construction, or below the floor at the ceiling beneath. Risers to upper floors should be run not over 2″ from the finished walls when they are exposed in the rooms.

By using *top flow* and *bottom return connections* at each radiator it is possible to positively control each unit by a single valve, except for the slight circulation intended to prevent freezing, which takes place through the 1/16″ diameter hole drilled in the valve disc or sleeve, when the valve is closed. If both connections are made at the bottom tappings, and only one valve is used, it is entirely possible that the radiator may *still* be supplied with hot water through the unvalved connection even when the valve is closed.

Air Removal in Hot-Water Systems. Suitable provision must be made for the removal of air from all hot-water radiators, wherever an up-feed system is installed. Usually small air

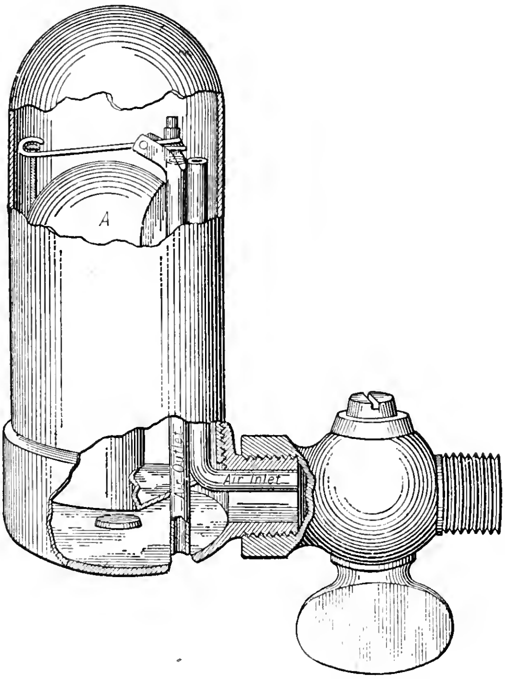


FIG. 9. MONASH HOT-WATER AIR VALVE.

cocks are attached to the highest point of each radiator and are periodically opened to relieve any accumulation of air. If these cocks are forgotten a radiator may become air bound and fail to heat due to faulty circulation, hence automatic air valves are sometimes installed for this purpose.

The automatic air valve for hot-water radiators is not very generally used, due to its liability to pass water as well as air, but a standard type, made by the *Monash-Younker Co.*, is shown in Fig. 9. This valve is attached at the high point of the radiator and whenever air accumulates in same the float *A* will drop and uncover the upper end of the air outlet tube permitting the air to escape until the returning water lifts the float and closes the outlet. This process is repeated periodically as air collects in the radiator and passes into the valve. A shut-off cock is provided to avoid accidents and damage should the valve leak water.

EXPANSION TANKS FOR HOT-WATER HEATING

Open-Tank Systems. The low pressure system of hot-water heating is not a closed system, as provision must be made for expansion and contraction of the water within the system. An open tank is provided at a suitable elevation, not less than 3 feet above the highest radiator, and connection made to the nearest return riser, or preferably a separate expansion line is run to the flow or return main in the basement.

The *size of expansion tank* varies with the amount of water in the system, and also with the range in temperature of same, and its capacity is determined as follows:

The increase in volume of a given weight of water heated from 32° to 212° is about 1/23 or approximately 4.33%, so that for every 23 gallons in the system at 32° an allowance of 1 gallon must be made in the expansion tank, when the water in the system is raised to 212°. Cast iron radiators have an internal volume of 1½ pints per sq. ft., while steel radiators and 1″ pipe hold about 1 pint per sq. ft.

Assuming the internal volume of the radiators is about 50% of the entire system we have for 3,000 sq. ft. of actual radiation, $3000 \times 2 \times 1/8$ gals. = 750 gals. of water. This water will increase $1/23 \times 750 = 33$ gals. on being heated from 32° to 212° Hence an expansion tank of $2 \times 33 = 66$ gals. capacity is necessary, the tank being made double the theoretical volume for practical considerations.

A list of expansion tank capacities and dimensions is given in Table 7, from which a commercial tank may be readily selected for systems under 6,000 sq. ft. For larger systems the size of tank should be separately determined and the nearest commercial tank size, as taken from manufacturer's list, should be specified. These tanks should have 1″ or 1¼″ top and bottom tappings with ½″ water gage tappings, for connecting a gage glass at least 12″ long on the side of tank as shown in Fig. 11.

The tank must be securely supported well above the highest radiator in the system, and in the larger installations special framing must often be designed to carry the weight of tank and water.

Automatic expansion tanks equipped with a ball cock and overflow are sometimes installed, and the altitude gage on boiler, and the gage glass and fittings on tank omitted. These tanks may be covered with hardwood and varnished if it is necessary to place them in a finished room or apartment.

Expansion Tank Connections. The most approved method of connecting an expansion tank to a low pressure one-pipe system is shown in Fig. 11, where an expansion and vent line is run from the top of the main, at the high point just above the boiler, and is connected to a return bend just beneath the tank. A return circulating line is taken from the other side of this bend and connected into the return main at the boiler. The circulation of water in this loop will prevent freezing at the tank. From the top of the tank a 1¼″ vent line is taken through the roof, and a 1¼″ overflow is taken out of this vent line at a tee just above the tank. This overflow should discharge into an open sink or drain near the boiler so that it will be immediately evident to a person in the boiler room, filling the system, just when the water has risen to the overflow above the tank. The movable hand on the boiler altitude gage can then be set to correspond with the middle of the gage glass, and the water level brought to this point with the system cold.

No valves should ever be installed *on either the expansion or the overflow lines*, and in case the system is valved at the boiler the expansion line must be connected on the boiler side of this valve, and where two boilers are installed this line must be carried to a point above the water line in the expansion tank to prevent siphoning the water out of the entire system should it be necessary to drain only one boiler.

Expansion or vent pipe connections must always be so made to main piping in basement that all air will be automatically removed from high points. Wherever possible risers or branches to risers may be used for relieving any accumulation of air in the main piping.

HOT WATER HEATING – EXPANSION TANKS
(OPEN AND CLOSED SYSTEMS)

TABLE 7

Expansion Tanks

No.	Size in Inches	Capacity Gal.	Sq. ft. of Radiation
0	10 x 20	8	250
1	12 x 20	10	300
2	12 x 30	15	500
3	14 x 30	20	700
4	16 x 30	26	950
5	16 x 36	32	1300
6	16 x 48	42	2000
7	18 x 60	66	3000
8	20 x 60	82	5000
9	22 x 60	100	6000

NOTE – Galvanized Steel, tested at 100 lb. Tapped $1\frac{1}{4}$" top and bottom, and for gage glass.

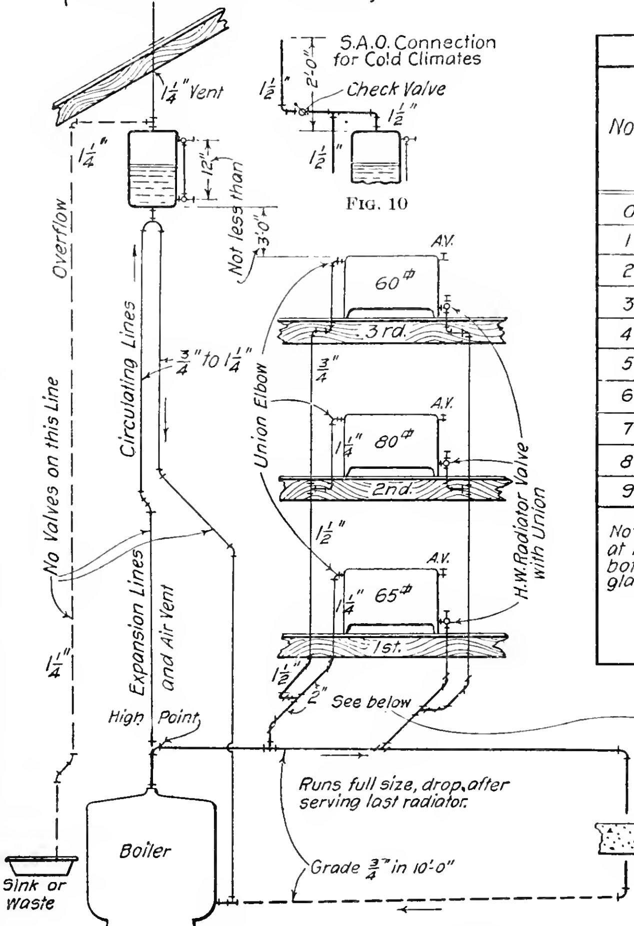
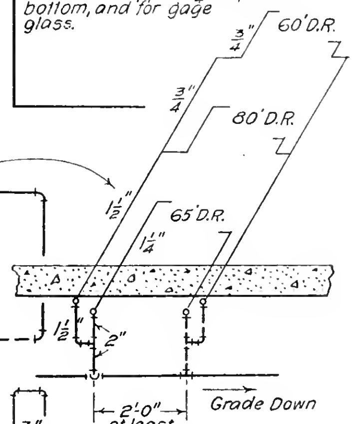
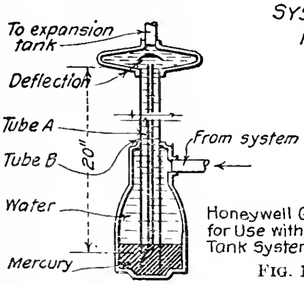


FIG. 10

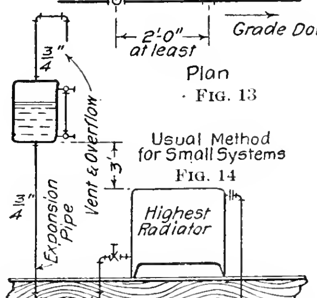
ONE PIPE UNDERFEED SYSTEM – OPEN TANK
For Piping Sizes see
TABLES 1 AND 2
FIG. 11



Plan
FIG. 13



Honeywell Generator for Use with a Closed Tank System
FIG. 12



Usual Method for Small Systems
FIG. 14

In *small installations* the expansion line may be connected to the return riser of one of the highest radiators, and no overflow other than the vent need be provided for, as shown in Fig.14. This is a cheap method, and should not be resorted to unless extreme economy must be practised. The tank must be in the same room with the radiator to prevent freezing, as no circulation is provided for, and the overflow is simply discharged out of doors and usually upon the roof. The usual result is that an unsightly appearance is soon created.

The *U. S. Treasury Dept.* employs a special vent and overflow connection (Fig. 10) in cold climates, where there is a liability of the vent line freezing up if run out through the roof, due to the condensation and freezing of vapor passing out through this line. The vent line is made only 2′–0″ high above the tank so that it is kept within the building, and is equipped with a check valve to prevent the escape of water through same in case the tank should overflow suddenly. The closing of the check valve will compel the excess water to pass down the overflow, and prevent flooding of the building.

Closed Tank Systems. The *permissible temperatures* in any hot water system are limited by the pressure on the system, which latter factor determines the point at which boiling will take place.

The pressure at any elevation in an *open tank* hot water system will vary directly with the distance below the level of the water in the expansion tank, and hence it will be possible theoretically to carry the water in the boiler at a temperature corresponding to the hydrostatic pressure at the boiler before boiling would occur. The relation between hydrostatic head, pressure, and boiling point are given in the following table:

TABLE 8

RELATION BETWEEN HYDROSTATIC HEAD, PRESSURE, AND BOILING POINT

Hydrostatic Head in Feet...	0	12	24	37	49	61	74	87	100	113	125
Pressure in Lb. per Sq. In....	0	5	10	15	20	25	30	35	40	45	50
Boiling Point.............	212	227	239	250	259	268	274	281	287	292	298

Practically it would be quite impossible to carry temperatures in excess of 212° in any part of an open tank system, as the high temperature water would immediately rise into the open tank and boil.

In order to overcome the limitations of the open tank system, wherein boiling will always occur as soon as 212° in temperature is reached, various means of increasing the pressure in these systems have been resorted to in the attempt to carry a higher water temperature in the radiators in very cold weather than would be possible with an open-tank system. These devices have usually been installed on the expansion line either at the boiler or else just below the expansion tank, and increase the static head by interposing a column of mercury in the path of the expanding water as it flows into the expansion tank.

A common form of the apparatus, known as the *Honeywell Heat Generator*, is shown in Fig. 12 from which it will be seen that water entering the generator from the system will force the mercury up the inner tube *A* until a head of 20″ or 10 lb. is established, at which time the entrance to this tube will be uncovered by the mercury and water or air may enter same and pass to the expansion tank. Any excess of mercury above that required to just fill tube *A* is returned by tube *B* to the reservoir in the base. When the system cools off water can flow back down tube *A* as soon as the mercury column drops in same, and the slight head of mercury then existing at the outlet of this tube is easily overcome by the head of water in the expansion tank above this point.

This increase of 10 lb. in static pressure makes it possible to carry a maximum water temperature of 240°, or nearly 30° higher than would be possible in an open tank system. While a temperature as high as this could theoretically be carried *at the boiler* in an open tank system with a static head of 24 feet, just as soon as this water rose in the system it would boil, and

escape from the expansion tank, at the same time emptying the system of water. In fact with the open tank system the water is liable to be driven out at temperatures of 212° F.

The use of pressure generators similar to the above makes it possible to use smaller radiators in the heated rooms, as it is entirely possible to maintain steam temperatures in the radiators whenever desired. Since higher temperatures are used, the difference between flow and return riser temperatures becomes greater than in the open tank system and hence a greater motive head exists so that smaller mains and risers may be used with this system. The *Honeywell Company* recommends the following schedule of radiator tappings:

TABLE 9

RISER SIZES FOR HONEYWELL SYSTEM

Pipe Size	Capacity in Sq. Ft. of Hot Water Radiation		
	1st Floor	2nd Floor	3rd Floor
In.			
½	30	40	50
¾	75	100	125
1	75 up	100 up	125 up

It should be remembered that since radiators and pipes are smaller in this system there is much less water than in the open tank system, making it more sensitive in warming up and *also in cooling off.*

The "generator" should not be placed close under the expansion tank, otherwise than this its location may be anywhere in the expansion line, as the same hydrostatic head is always acting in addition to the head of mercury column.

PIPING SIZES FOR GRAVITY HOT-WATER HEATING

The basis upon which the piping sizes are proportioned in hot water heating must be, as in the case of steam heating, the number of square feet of radiation to be supplied, or, more exactly, the actual number of B.t.u. per hour required by this radiation.

The number of B.t.u. required is, $H = K(t_w - t_a) \times R$, or $R = \dfrac{H}{K\ (t_w - t_a)}$ where R = sq. ft. of direct radiation, H = B.t.u. loss to be supplied by the radiator per hour, K = coefficient of transmission for the radiator, t_w = *average* temperature of water in the radiator, and t_a = temperature of surrounding air. In practice the radiation is proportioned on the assumption that the water enters the radiator at 180° F. and leaves at 160°, giving a 20° drop with an average temperature of 170° for maximum conditions. By using a greater drop, say from 180° to 144°, less water per hour or lower velocities of flow may be used, as will be shown later.

It is therefore evident that if the heat loss H is known, and the quantity of water in cu. ft. to be supplied per hour is Q, then $H = Q \times D \times 20$, where D = density of water per cu. ft. at 170°, = 60.77 lb., or $Q = \dfrac{H}{20 \times D}$, or $Q_m = \dfrac{H}{1200 \times D}$ per minute. Hence to supply one sq. ft. of radiating surface having a transmission rate of 150 B.t.u. per hr. we need $150/20 = 7\frac{1}{2}$ lb. or $150/(20 \times 60.77) = 0.123$ cu. ft. of water at 170°, or $(0.123 \times 1728)/231 = 0.92$ gals. per hour.

From which it follows that the area of the pipe required in sq. ft. is, $A = Q_m/V$, where V = velocity in ft. per minute. Conversely the velocity required for any particular radiator can be determined as follows: $V = Q_m/A = H/(A \times 1200 \times D)$ ft. per minute. Now if d = diam. of pipe in inches, and V_s = velocity in feet per second we have,

$$V_s = \frac{H}{\frac{\pi d^2}{4} \times \frac{1}{144} \times 1200 \times 60 \times 60.77}$$

V_s varies from 0.25′ to 1′ per sec. for small pipes, and from 0.5′ to 2′ per second for large mains of 12″ diameter.

Now since Q is definitely fixed for each case the size of the pipe will depend entirely on the velocity with which the water will flow through the system.

Theoretical Velocity of Flow (No Friction). The velocity of flow in gravity hot-water heating depends entirely on the difference in weight or head of the water in the flow and return risers, provided there is no friction; the water in the latter being denser or weighing more per cu. ft. than the water in the flow riser. This difference in weight or head is the actuating force producing flow, and the theoretical velocity it will produce can be found by substituting the proper values in the velocity formula, $V = \sqrt{2ah}$, where V = velocity in feet per sec., a = acceleration in ft. per sec.² due to this force, and h = head in feet of the fluid flowing.

The relationship between the force acting and acceleration produced in a hot-water system can be best understood by reference to Fig. 15, where two weights, A and B, are shown connected by a weightless cord hung over a frictionless pulley. As A is heavier than B, the actuating force is $A - B$ and the weight moved is $A + B$, so that $a : g = A - B : A + B$, the moving mass being the same whether the force $(A - B)$ or the force of gravity $(A + B)$ acts on the system. From which it follows that $a = g\left(\frac{A - B}{A + B}\right)$ and hence the theoretical velocity of the system moving through the height h is, $V = \sqrt{2g\left(\frac{A - B}{A + B}\right) h}$.

The application of this formula to a hot-water problem is shown in Fig. 18, where an open tank system is shown. W_1 = density per cu. ft. of water in the flow riser, and W_2 = density of water in the return riser, h = elevation of radiator above boiler, center to center, V = velocity in ft. per sec., and g = 32.2. Then $V = \sqrt{2gh\frac{(W_2 - W_1)\ h}{(W_2 + W_1)\ h}}$ and substituting we have $V = \sqrt{2 \times 32.2 \times 10 \times 0.00354}$ = 1.55 ft. per sec. for 10′ head and 20° drop between flow and return risers.

The following table by *R. C. Carpenter* gives the theoretical velocities in hot-water pipes in feet per second for various temperature differences between the flow and return with varying heads:

TABLE 10

THEORETICAL VELOCITY IN FEET PER SECOND IN HOT-WATER PIPES

(h) Height of System in Feet	Free Fall in Air*	Difference of Temperature ($t_1 - t_2$)						
		1°	5°	10°	15°	20°	30°	40°
1	8.03	0.107	0.242	0.335	0.412	0.478	0.593	0.672
5	17.90	0.232	0.541	0.750	0.922	1.090	1.330	1.510
10	25.40	0.328	0.765	1.060	1.320	1.550	1.880	2.140
20	35.90	0.463	1.085	1.500	1.850	2.190	2.660	3.010
30	43.90	0.567	1.330	1.830	2.260	2.680	3.260	3.710
40	50.70	0.656	1.530	2.120	2.610	3.080	3.760	4.260
50	56.70	0.732	1.710	2.370	2.820	3.470	4.220	4.770
60	62.10	0.802	1.880	2.590	3.200	3.790	4.620	5.220
70	67.10	0.866	2.020	2.800	3.450	4.080	4.970	5.650
80	71.80	0.925	2.160	3.000	3.690	4.370	5.320	6.030
90	76.80	0.932	2.290	3.180	3.910	4.640	5.640	6.410
100	80.30	1.037	2.420	3.350	4.130	4.780	5.920	6.720

* NOTE.—Velocity acquired by body falling through the distance h.

HOT WATER HEATING - GRAVITY SYSTEMS
HEAD PRODUCING FLOW

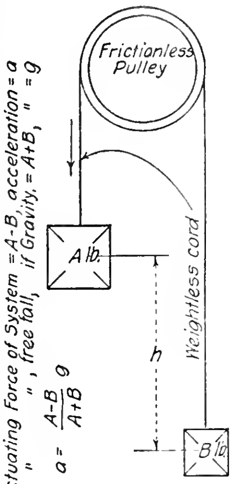


FIG. 15

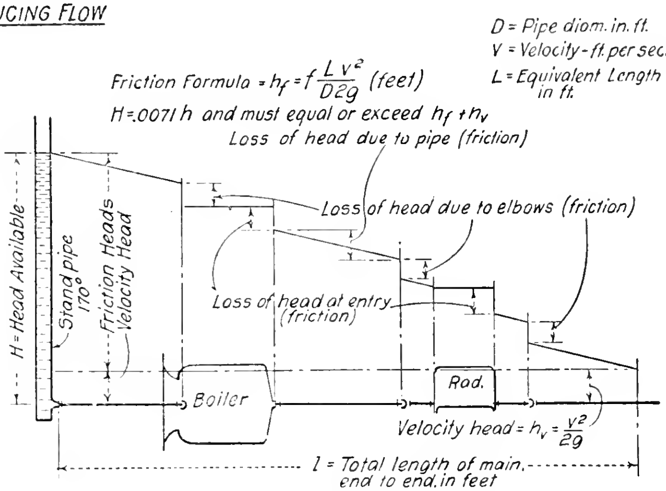


Diagram Showing how Head Available is Balanced against Friction and Velocity Head

FIG. 16

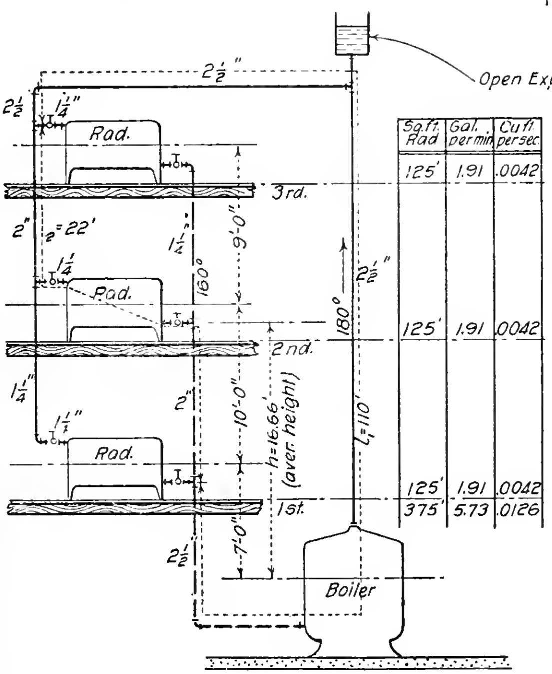


FIG. 17

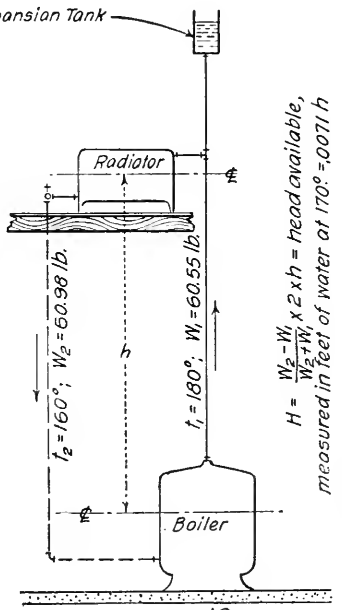


FIG. 18

Actual Velocities of Flow. Actual tests show from 20 to 50% of the tabulated or theoretical velocities, due to friction losses in the pipe line.

A calculation for the velocity of flow in a gravity system is shown herewith. $H \times \frac{W_2 + W_1}{2} \times$ area pipe $= h \times (W_2 - W_1) \times$ area pipe in sq. ft. See Fig. 18. $H = h\left(\frac{W_2 - W_1}{W_2 + W_1}\right) \times 2 =$ 0.0071 h feet, measured at 180° and 160° with H taken at the mean temperature of the flow and return risers. H = head in feet producing flow, and h = mean height of system.

It will be apparent that the actual velocity of flow for any given system can be readily found, provided the flow and return temperatures are known, and then the ratio of the actual to the theoretical velocity easily determined. In no case should it be greater than unity.

Such a system is shown in Fig. 17, and all radiator and piping sizes, as well as lengths of circuits, elevations and temperatures are fully indicated. The average height of the system is

$$h = \frac{125 \times 26 + 125 \times 17 + 125 \times 7}{375} = 16.66'.$$

The total volume of water flowing per hour is, $\frac{375 \times 150}{20 \times 60.8} = 46.1$ cu. ft. at 170° or 46.1 / 3600 = 0.0128 cu. ft. per second. The internal diameter and area of 1¼″ pipe is, 0.115′ and 0.0104 sq. ft.; of 2″ pipe, 0.178′ and 0.0233 sq. ft.; and of 2½″ pipe, 0.205′ and 0.0332 sq. ft. respectively.

Hence the velocity in the 1¼″ branches to each radiator is $\frac{Q_s}{A} = 0.0042/0.0104 = 0.404$ ft. per sec. and in the main riser it is 0.0128/0.0332 = 0.386 ft. per sec. In the same way the velocity in the 2″ risers is found to be 0.0085/0.0233 = 0.365 ft. per sec.

Now by reference to Table 10 at a temperature difference of 20°, and at a head of 16.66′ the theoretical velocity is, by interpolation, about 2′ per sec., so that the maximum velocity in this system is only $\frac{0.404}{2} \times 100 = 20.2\%$ of the theoretical.

The head available for producing flow in the system (Fig. 17), as already given above, is $H = 0.0071\,h = 0.0071 \times 16.66 = 0.118$ ft. of water column at 170°, or $0.118 \times 60.77 = 7.16$ lb. per sq. ft., which must be equal to, or greater than, the sum of the velocity head plus all the resistances in the entire circuit. See discussion following under "Use of Friction Pressure Loss Chart," for method of selecting pipe sizes when head or pressure available and heat loss to be supplied are known.

Practically, the *total head H*, in feet, available for producing flow (difference in height of two columns of water at temperatures t_1 and t_2 when in equilibrium) must equal the sum of the three heads to be overcome.

Heads to be Overcome. The three heads to be overcome in any gravity hot-water system, and their relation to the head $H = 2\left(\frac{W_2 - W_1}{W_2 + W_1}\right) h$ available for producing flow are shown diagramatically in Fig. 16, where it will be seen that the head available is just equal to the sum of all the lost heads plus the velocity head.* These heads may be classified or distinguished as follows:

(1) *Velocity head*, or height a body must fall in a vacuum to acquire the velocity of flow in the pipe line, expressed in feet of fluid flowing.

* NOTE.—The velocity head, which is extremely small with a gravity system, must only be overcome at the time the system is started.

(2) *Entry head,* or head needed to overcome resistance at entrance to flow pipe at boiler and return at radiator, expressed in same units as above.

(3) *Friction head,* or head required to overcome resistance to flow within the pipe. For any given velocity and diameter of pipe the friction head increases directly with the length. For a given length the friction head increases with the velocity and decreases as the diameter grows larger. This is not a simple relation, as in the case of change of length only, but varies as some function of the velocity and diameter. See exact equations following as given by *Dr. R. Biel* for friction head.

It is customary to include entry head under friction head by the use of a suitable factor, in the same way that fittings and valves are allowed for in terms of equivalent length of straight pipe.

Due to the fact that velocities are very low in gravity work it is usually customary to neglect the velocity head in computations, and design piping of such size that the head available will always exceed the friction head, including the entry head.

The typical and usual formula for the friction pressure loss of water flowing in a pipe line is,

$$h_f = f \times \frac{L}{D} \times \frac{V^2}{2g},$$

and as already stated h_f = head in feet of water flowing, f = a coefficient of friction, L = equivalent length of pipe in feet, D = diameter in feet, V = velocity in feet per second, and $g = 32.2$. This formula, derived by *Weisbach,* using friction coefficients f, determined by him or later by *Chezy,* is true only for relatively high velocities of flow and cold water, so that its application to gravity hot-water heating with very low velocities and high temperatures will give, generally, approximate or unsatisfactory results.

More Exact Equations for Friction Pressure Loss. Recent investigations by *Dr. R. Biel* on both hot and cold water, and published in *Zeitschrift des Vereins Deutscher Ingenieure,* 1908, indicate that *a single simple formula cannot be used for the friction head loss due to flow of hot water in pipes,* but that at least three formulas are necessary to express the relation. The formula to be used depends on the velocity of flow, and two "critical velocities" have been experimentally determined so that the selection of the proper formula, (A), (B) or (C) may be made; using (A) for velocities below $V_1 = 0.158/d$, *the lower critical velocity,* (B) for velocities between V_1 and $V_2 = 1.382/\sqrt{d}$ *the upper critical velocity,* and (C) for velocities above V_2. Both V_1 and V_2 are in feet per second and d = nominal pipe diameter in inches.

The three friction formulas, as deduced from the experimental data, are as follows:

(A) for velocities below V_1; $h = 0.0111 \times \dfrac{L \times V}{d^2}$ or $P = 0.05643 \times \dfrac{L\,V}{d^2}$;

(B) for velocities between V_1 and V_2; $h = \dfrac{LV^2}{d} \times 0.1757\left((0.33 + \dfrac{0.226}{\sqrt{d}}\right)$

$$\text{or } P = \frac{LV^2}{d} \times 0.893\left(0.33 + \frac{0.226}{\sqrt{d}}\right);$$

(C) for velocities above V_2; $h = \dfrac{L\,V^2}{d} \times 0.1757\left(0.12 + \dfrac{0.226}{\sqrt{d}} + \dfrac{0.288}{V\sqrt{d}}\right)$

$$\text{or } P = \frac{L\,V^2}{d} \times 0.893\left(0.12 + \frac{0.226}{\sqrt{d}} + \frac{0.288}{V\sqrt{d}}\right)$$

Both the influence of temperature and the change in viscosity of hot water are allowed for in the formulas of "critical velocity" and friction head, as given above, and the friction for-

mulas hold true for the circulation of water in iron pipes such as are commonly used for domestic engineering purposes.

In each of the above formulas h = friction head at the mean temperature of flow and return in inches of water column, L = length of pipe in feet, d = internal diameter of pipe in inches, V = velocity of flow in feet per second, and P = pounds per sq. ft. In changing from inches of water to pounds the density of water has been taken as 60.94 = 61 lb. per cu. ft. at the mean of the flow and return line temperatures or $(180 + 144)/2 = 162°$ F. Density at 170° = 60.77 lb. This allows for a drop of 36° F. in the system, and the relation between velocity in feet per second and pipe diameter to supply a given number of B.t.u. per hour is easily derived as follows: Area of pipe in sq. ft. × velocity in feet per hour × density of water = B.t.u. per hr., and hence,

$$V = \frac{\text{B.t.u. per hour}}{3600 \times 36 \times 61 \times (d/12)^2 \times 3.14 \times \frac{1}{4}} = \frac{\text{B.t.u. per hour,}}{43{,}096. \times d^2} \quad . \quad . \quad . \quad . \quad . \qquad (D)$$

where d = the diameter of the pipe in inches, and V = velocity in feet per sec. For a drop of less than 36° in the water temperatures higher velocities must be used for the same number of B.t.u. per hour and pipe diameter. Thus if only 20° drop in temperature is to be allowed the velocity would be 36/20 or 9/5 of that determined above.

Application of the Formulas and Construction of Chart. The rapid and practical application of the friction formulas given above to general practice over a wide range of friction losses, pipe diameters and velocities can best be provided for by showing these relations graphically on logarithmic plotting paper as laid out on the chart Fig. 19.

This chart, as well as the manner in which it was constructed and the method of using it, is taken from an article by *I. V. Serginsky* in the *Heating and Ventilating Magazine* for Nov., 1913.

The chart (Fig. 19) gives directly the friction pressure loss (axis of ordinates) in pounds (ranging from 0.001 to 1.0 lb.) per sq. ft., per running foot of flow and return pipe required to supply from 1,000 to 10,000,000 B.t.u. (axis of abscissæ) per hour, by water flowing in pipes from ½″ to 12″ in diameter.

Lines showing velocity of flow are also laid out on the chart, all curves being based on a flow temperature of 180° and return at 144°, which gives a total drop of 36° per 1 lb. of water circulated.*

The manner in which the formulas for critical velocity, friction pressure loss, and actual velocity were used in plotting the chart is illustrated by taking a specific case.

Given a straight pipe 1½ inches diameter and 1 ft. long, then:

V_1 = 0.105 ft. per sec. lower "critical" velocity

V_2 = 1.129 ft. per sec. upper "critical" velocity

From equa. (A), $P_1 = 0.02505 \times V$ lb. per sq. ft. per 1 ft.

From equa. (B), $P_2 = 0.3063 \times V^2$ lb. per sq. ft. per 1 ft.

From equa. (C), $P_3 = 0.595 \times V^2 \times \left(0.304 + \frac{0.235}{V}\right)$ lb. per sq. ft. per 1 ft.

From equa. (D), $V = \frac{\text{B.t.u. per hour}}{96{,}966}$ ft. per sec.

At the lower critical velocity $P_1 = 0.025 \times 0.105 = 0.0026$ lb. per sq. ft. per running foot, and at the upper critical velocity $P_2 = 0.31 \times 1.13^2 = 0.391$ lb. per sq. ft. per running foot. From the values of P_1 use equation (A) up to 0.0026 lb. per sq. ft., and from P_2 use equation (B) up to 0.391 lb. per sq. ft. and for a greater friction head loss than 0.391 lb. use equation (C).

*Note.—For other temperature drops than 36°, say 20°, use the same chart, but take 20/36 or 5/9 of the values now plotted as abscissæ.

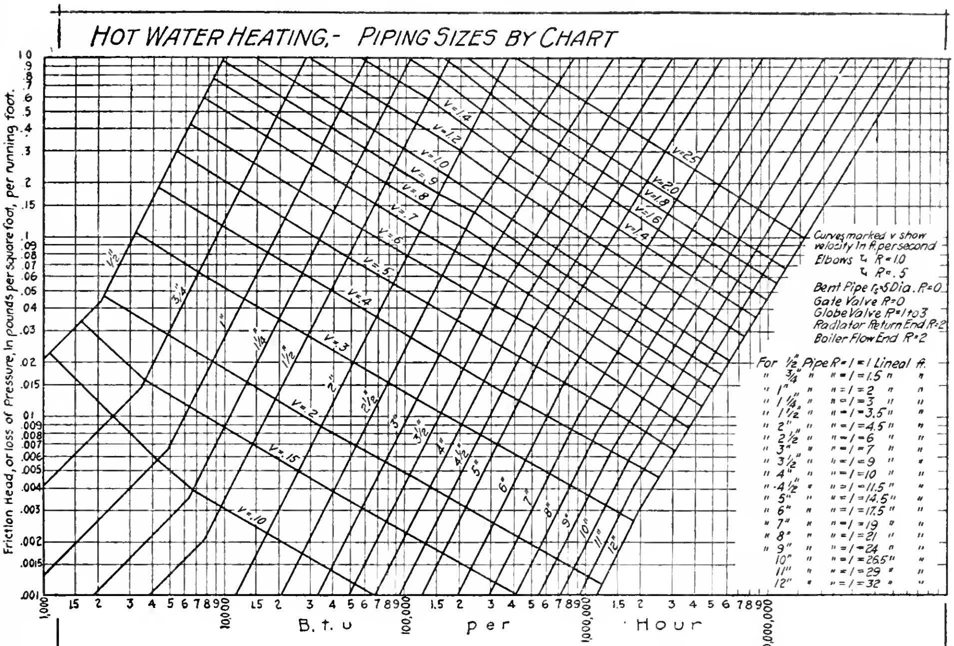


FIG. 19. Friction Pressure Loss Chart for Gravity Hot Water Heating Systems.

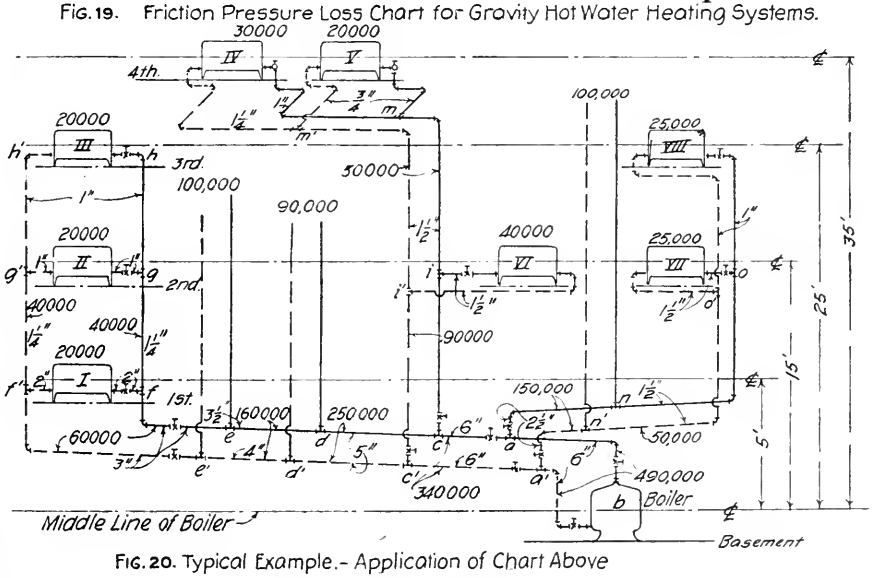


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FIG. 20. Typical Example.- Application of Chart Above

Then at 0.001 lb. pressure loss on the chart the velocity of flow and B.t.u. per hour are found as follows by equations (*A*) and (*D*). $0.001 = 0.025 \times V$ or $V = 0.04$ and also $V \times 96{,}966$ = B.t.u. per hr. = 3,878, and again at $0.002 = 0.025 \times V$ we have $V = 0.08$ and also $V \times 96{,}966$ = B.t.u. per hr. = 7,757, which determines two points on this line. Since there

is no change in the shape of the formula below 0.0026 lb. pressure loss this part of the line for $1\frac{1}{2}''$ pipe is straight and can at once be drawn in. A similar calculation at 0.004 and 0.010 lb. pressure loss, using equations (*B*) and (*D*), gives $0.004 = 0.306 \times V^2$ or $V = 0.114$, and also $V \times 96{,}966$ = B.t.u. per hr. = 11,050. Also at 0.010 lb. the B.t.u. = 17,453 per hr. Plotting these last two points in B.t.u. against 0.004 and 0.010 on the axis of ordinates we get another straight line intersecting the first line (already located) at a small angle as shown on the chart for $1\frac{1}{2}''$ diameter of pipe.

In the same way formulas (*C*) and (*D*) should be used for friction pressures above 0.391,

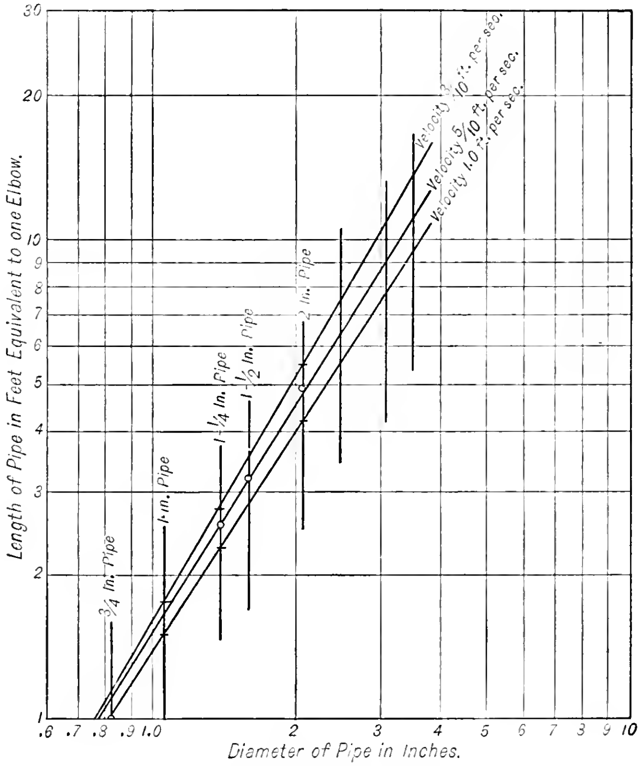


Fig. 21. Pipe Equivalent of 90° Elbow.

but as this was near the upper limit of the chart a third line was not calculated for $1\frac{1}{2}''$ pipe, corresponding to velocities above the upper "critical" velocity as was done for $2\frac{1}{2}''$ and larger pipe.

Allowance for Resistance of Fittings and Other Obstructions. It is possible to express the friction loss due to fittings as $H_f = R \times V^2/2g$ feet of water column, where V = velocity of flow in feet per second. Values of R taken as follows: Sharp bend = 1, short bend = 0.5, long bend (radius greater than $5d$) = 0, gate valve = 0, globe valve = 1 to 3, outlet from boiler or return end of radiator = 2 (vena contracta).

Now if the value of R for a given resistance is equal to one and we use the formula (*B*) for friction pressure loss in inches of water as it is the most generally applicable of the three formulas, we have:

$$R \times 12 \times \frac{V^2}{2g} = \frac{LV^2}{d} \times 0.1757 \left(0.33 + \frac{0.226}{\sqrt{d}}\right)$$

which is the relation between R, L and d, as V cancels out of the equation.

Also when $R = 1$, $L = \dfrac{12 \times d}{64.4 \times 0.1757 \left(0.33 + \dfrac{0.226}{\sqrt{d}}\right)}$ and for 1″ pipe, $d = 1$ and $L = 1.9$ or 2 ft.; while for 4″ pipe, $d = 4$ and $L = 9.6$ or 10 ft. Hence for a 4″ globe valve, where $R = 2$ the equivalent length of pipe is $2 \times 10 = 20$ ft., and for a 1″ short bend where $R = 0.5$ the

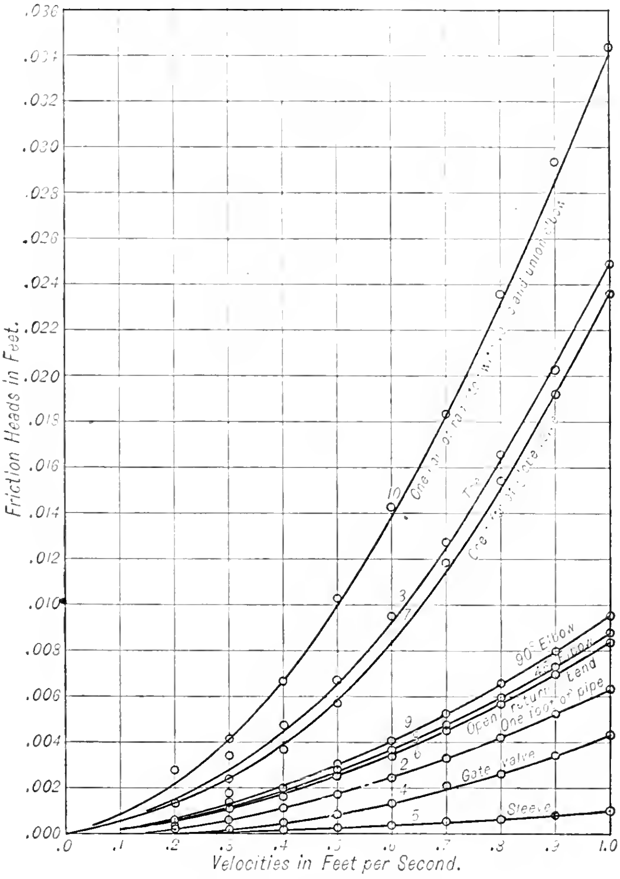


FIG. 22. COMPARATIVE FRICTION LOSS IN FITTINGS.

equivalent length of pipe is $0.5 \times 2 = 1$ ft. Again for 6″ pipe $L = 17.5$, and for entry loss at boiler outlet allow $R = 2$ or $2 \times 17.5 = 35$ ft. of straight pipe as equivalent. For values of L for $R = 1$ for other sizes of pipe, see the chart, Fig. 19.

The relation between the friction head and the velocity of water through pipe fittings and valves is also shown in the curves (Figs. 21 and 22), by *F. E. Giesecke*.

Use of the Chart and Its Application to an Actual Problem. As already stated, it is always necessary to make the sum of the various friction heads and the velocity head less than the "head available" for producing flow. Also the head available is the difference between the densities of the water in the flow and return risers, multiplied by the average height of the system in feet. Hence if we have two columns of water at 180° and 150° respectively and each 50′ high with a connection at the base, the head or force available is $(61.201 - 60.548) \times 50 = 0.653 \times 50 = 32.65$ lb. per sq. ft.

Now by reference to the diagram, Fig. 20, showing a two pipe hot water layout with basement mains, it will be seen that radiator *fIf′* is farthest away and at the lowest point, or 5′ above the center of the boiler, so that the available pulling force is 5×0.653 or 3.265 lb. per sq. ft.

TABLE 11

LENGTHS OF PIPING AND NUMBER OF VALVES AND FITTINGS

(See Fig. 20)

Section	ab	a′b	ac	a′c′	cd	c′d′	de	d′e′	ef	e′f′	fg	f′g′	fIf′	gIIg′	gh
Length feet	30	30	20	20	40	40	30	30	10	15	10	10	10	10	10
No. Globe Valves	1	1	1	1	..	..	..	..	1	1	..	..	1	1	..
No. Short Ells	4	4	1	1	1	1	2	2	2	2	..	..	8	8	..

Section	g′h′	hIIIh′	ci	c′i′	iVIi′	im	i′m′	mVm′	mIVm′	an	a′n′	no	n′o′	oVIIo′	oVIIIo′
Length, feet	10	10	10	10	40	20	20	10	50	10	10	20	20	10	25
No. Globe Valves	..	1	1	1	1	..	..	1	1	1	1	..	..	1	1
No. Short Ells	..	8	1	1	10	3	3	10	10	2	2	3	3	8	10

The total length of circuit (Table 11) is 285′, not including elbows, valves, etc., and as from 50 to 100%, or say 75%, must be added to the actual length of the average circuit to allow for these extra resistances we have an equivalent length of $1.75 \times 285 = 500$ ft.

The allowable friction pressure loss in this complete circuit is, as already determined, 3.265 lb., or per ft. of length it is $3.265 / 500 = 0.0065$ lb. The velocity head is so small that it has been neglected and the entire head available utilized for overcoming friction.

Now, by reference to the chart at 0.0065 lb. on the ordinate axis and at 20,000 B.t.u. on the axis of abscissæ we can locate a point lying between 1½″ and 2″ pipe for the radiator connections, which have been taken as 2″.

In a similar manner the main sizes can be found for each section between the boiler and this radiator *fIf′* on the general assumption that on this circuit the force available is 3.265 lb. per sq. ft., and is due to the pull of this radiator alone, each section being made of the proper size to supply the B.t.u.required by the radiators served by it, as shown below:

fe, f′e′, each 60,000 B.t.u. = 3-in. pipes.
ed, e′d′, each 160,000 B.t.u. = 4-in. pipes.
dc, d′c′, each 250,000 B.t.u. = 5-in. pipes.
ca, c′a′, each 340,000 B.t.u. = 6-in. pipes.
ab, a′b, each 490,000 B.t.u. = 6-in. pipes.

Having thus obtained the sizes of the mains, it is possible to figure out the exact amount of friction head or loss of pressure at the dimensions found, in order to see how correct the dimensions are. It may appear that some sizes are too big and could be made smaller. Variation in this respect can be made so long as the total friction head is kept equal to the available pressure head of 3.265 lb. per square foot.

The procedure will be as follows:

Having the B.t.u. and size pipe for a part of the piping, find on the chart the corresponding friction head, or loss of pressure, in pounds per square foot per running foot; then multiply this by the total equivalent length of that part of the piping. Add all the products thus obtained for all parts of the circuit in consideration and the sum will be the total loss of pressure to be balanced by the available pressure head.

Assume for globe valves $R = 2$, and for elbows $R = 0.5$. Also make the assumption that all valves are of the globe type. Then from the chart:

$$ab = 6\text{ in.} = 490{,}000\text{ B.t.u.};\ p = 0.006 \times [30 + (2 + 2 + 4 \times \tfrac{1}{2}) \times 17.5]$$
$$= 0.006 \times 135 = 0.810\text{ lb. per square foot.}$$

Proceeding in a similar manner for the other portions of the piping, we get a total friction head on the circuit of radiator fIf' of 3.041 lb. per square foot, which is less than the balancing pulling force of 3.265 lb. If greater accuracy is desired, the circuit could be choked somewhere on the line, making a part of it smaller. For instance, make pipe cd 3½″ in diameter instead of 4″ as suggested.

In doing this, the friction in this part of the circuit is increased from 0.216 lb. to $0.01 \times [30 + (2 \times \tfrac{1}{2} \times 9)] = 0.01 \times 39 = 0.39$ lb., and the total friction head of the circuit will become $3.041 - 0.216 + 0.390 = 3.224$ lb. against the available pressure head of 3.265 lb. Piping thus figured can be considered accurate enough and a proper circulation is assured.

Proceeding to the next radiator directly above ($gIIg'$) the vertical distance from its center to the center of the boiler is 15 ft. The pressure head is, therefore, $15 \times 0.653 = 9.8$ lb. per square foot. This should be expended along the total length of the circuit from the boiler and back again. However, according to the preceding calculations the mains from the boiler up to the points f and f' require $3.224 - 0.138 = 3.086$ lb. per square foot, consequently for the part $fgIIg'f'$, there remains $9.8 - 3.086 = 6.714$ lb. per square foot, which should be equally distributed along this part.

Its length amounts to 30 ft. Allowing 50% for ells and valves, we get a total equivalent length $30 \times 1.5 = 45$ ft., which gives us a loss of pressure per running foot of $6.714 / 45 = 0.15$ lb. per square foot.

From the chart we get:

$fg = 40{,}000$ B.t.u. = 1¼-in.; $p = 0.135 \times 10 =$ .... 1.35 lb. per sq. ft.
$f'g' = 40{,}000$ B.t.u. = 1-in.; $p = 0.435 \times 10 =$ .... 4.35 lb. per sq. ft.
$gg' = 20{,}000$ B.t.u. = 1-in.; $p = 0.11 \times [10 + (2 + 2 + 8 \times \tfrac{1}{2}) \times 2] = 0.11 \times 26 =$ 2.86 lb. per sq. ft.
The sum $P =$ .... 8.56 lb. per sq. ft.

The loss of pressure here is greater than the available head, which must not be. Taking the next larger dimension of the pipe $f'g'$, we get $f'g' = 40{,}000$ B.t.u. = 1¼-in.; $p = 0.135 \times 10 = 1.35$ lb. per square foot, and the total loss of pressure will be $1.35 + 1.35 + 2.86 = 5.56$ lbs., against the available pressure head of 6.714 lb. This satisfies the condition that the loss of pressure is less than the available head. To avoid too rapid a circulation, in this line install an adjusting (shield) valve on the return end of the radiator, or a disc with hole through same.

The next radiator is $hIIIh'$. The vertical distance from the center of this radiator to the center of the boiler is 25 ft., and the available pressure head $= 25 \times 0.653 = 16.3$ lb. per square foot, which must be expended equally along the circuit from the boiler and back again. But according to the foregoing calculation, the friction head from the boiler along the piping up to the points g and g' amounts to $3.086 + 2.70 = 5.786$ lb. per square foot. For the part $ghIIIg'h'$ there remains, consequently, $16.3 - 5.786$ lb. $= 10.514$ lb. per square foot, to be distributed along the actual length of 30 ft., or $30 \times 1.5 = 45$ ft. equivalent. That gives $10.514 / 45 = 0.23$ lb. per square foot per running foot.

From the chart we get either:

1. $ghIII$ = 20,000 B.t.u. = 1 in.; $p = 0.1075 \times [10 + 5 + (2 + 4 \times \frac{1}{2}) \times 2] = 0.1075 \times 23 = 2.48$ lb. per square foot. $IIIh'$ = 20,000 B.t.u. = $\frac{3}{4}$-in.; $p = 0.48 \times [5 + (2 + 4 \times \frac{1}{2}) \times 1.5] = 0.48 \times 11 = 5.28$ lb. per square foot. $h'g'$ = 20,000 B.t.u. = 1-in.; $p = 0.1075 \times 10 = 1.08$ lb. per square foot. The sum $P = 8.84$ lb. per square foot.

2. $ghIIh'g'$ = 20,000 B.t.u. = 1-in.; $p = 0.1075 \times [30 + (2 + 2 + 8 \times \frac{1}{2}) \times 2] = 0.1075 \times 46 = 4.95$ lb. per square foot, or,

3. gh = 20,000 B.t.u. = 1-in.; $p = 0.1075 \times 10 = 1.08$ lb. per square foot. $hIIh'$ = 20,000 B.t.u. = $\frac{3}{4}$-in.; $p = 0.48 \times [10 + (2 + 2 + 8 \times \frac{1}{2}) \times 1.5] = 0.48 \times 22 = 10.55$ lb. per square

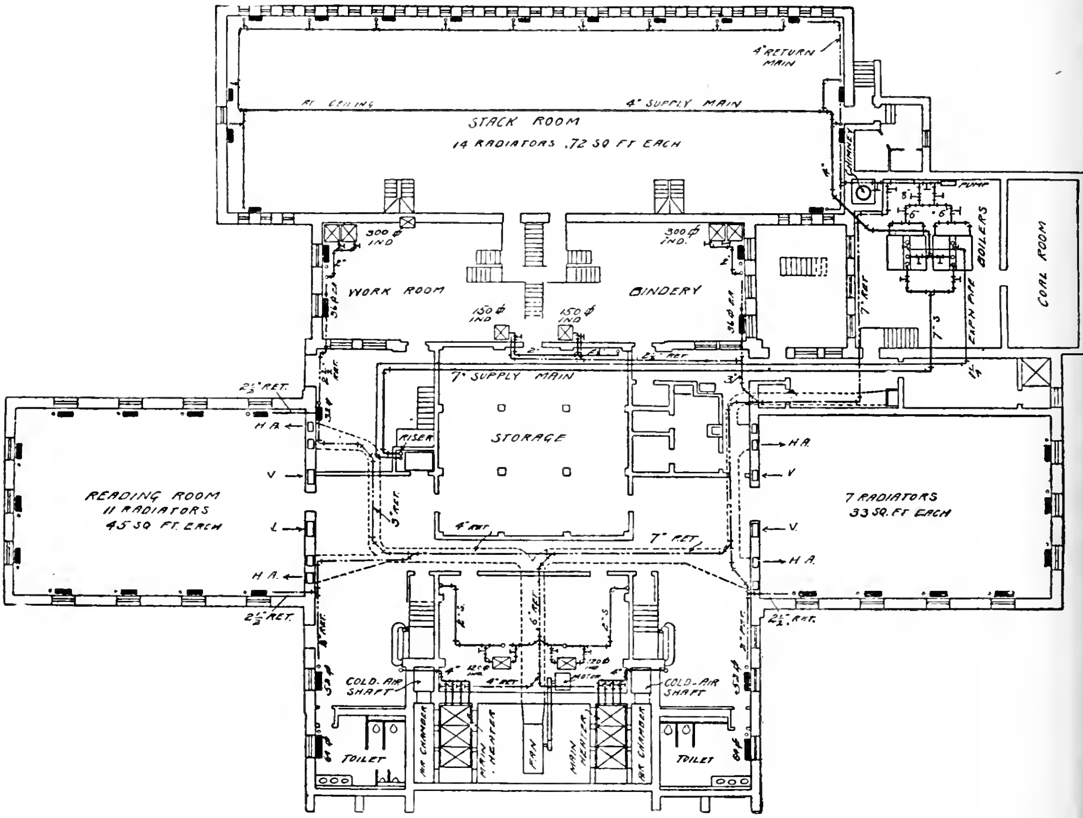


FIG. 23.

foot. $h'g'$ = 20,000 B.t.u. = 1-in.; $p = 0.1075 \times 10 = 1.08$ lb. per square foot. The sum $P = 12.71$ lb. per square foot.

Here only cases 1 and 2 can be taken into consideration, because in the third case the loss of pressure exceeds the available pressure head. In case 1 or 2, a shield valve must be employed, and the former case more nearly satisfies the conditions.

The loss of pressure for the other portions of the system may be figured in the same manner as for the circuits here worked out.

The method described in the foregoing of figuring a gravity hot water heating system is one which is based on correct scientific principles and is as exact as it is possible to make it. The

velocities at which water flows in gravity hot water heating apparatus can be obtained from the curves on the chart marked "v" which were determined by the use of equation (D).

Hot Water Heating System for a Library Building. A gravity circulating hot-water system, supplying both overhead and underfeed mains with an auxiliary pump and motor is shown in Figs. 23 and 24. This building, the Public Library of Brookline, Mass., is heated throughout by direct radiation, supplemented by indirect heaters for the book room and delivery room on the first floor and at the main entrance. The various reading rooms and the delivery room are supplied with fresh air by means of a centrifugal fan with belted motor in the basement. This

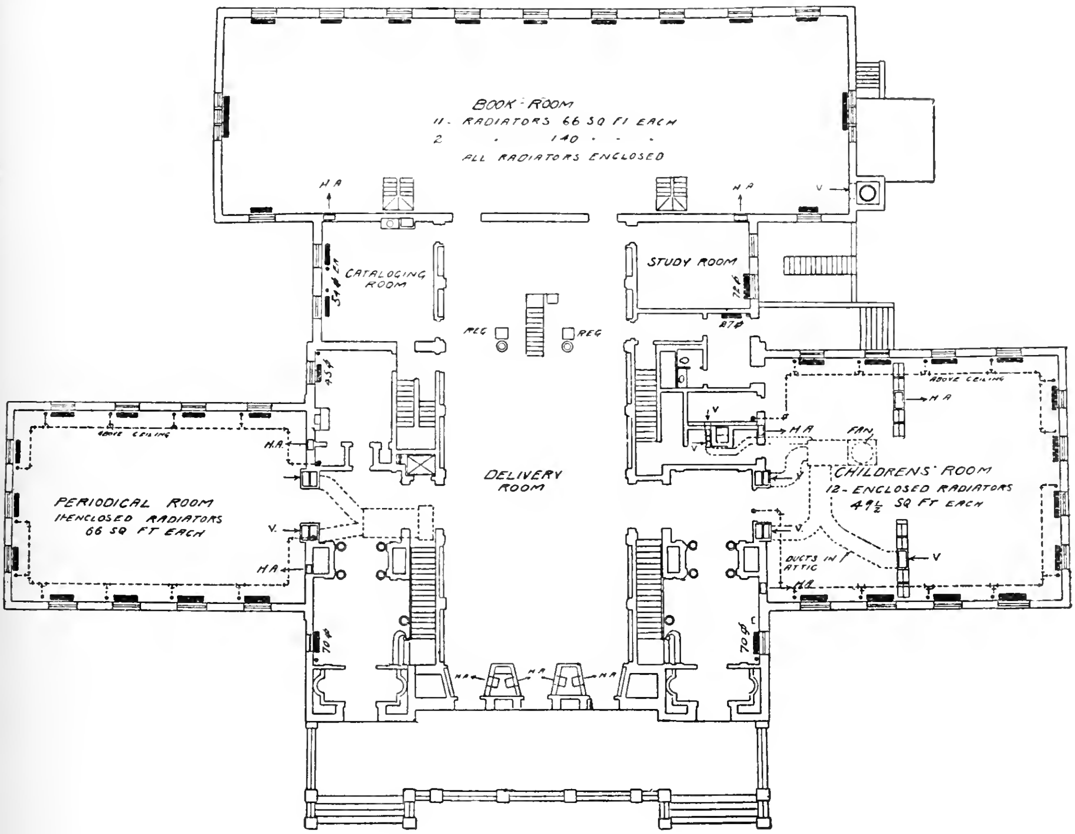


FIG. 24.

fan takes its air from the top of the building through two down-takes, one at each side of the main entrance as shown on first-floor plan. The right and left wing are vented through roof as shown, and the former is equipped with a vertical shaft disc, fan of the propeller type.

The 7" heating main for the central portion of the building, including the wings, is taken from the front of the headers on the two sectional cast-iron *Mills* boilers, and carried across the basement ceiling, as shown, to the foot of the elevator shaft, where it rises to the attic and supplies all the radiators in this part of the building by drop risers. The main returns are also shown, and are run in trenches or underground airways, as indicated.

The 4" heating main for the stack room is taken from the header just over the middle of

the boilers, and runs along the ceiling of the upper stack room. This main is arranged to feed the radiators in the book room above and the wall radiators in the stack room below, the same riser connection, taken from the side of the main, serving to supply the radiator above, and those in the stack room below, before entering the return which runs in a trench in the lower stack room floor. Practically all the drop risers are arranged to provide for a circulation at all times, whether the radiators are turned on or not, and hence are of the one-pipe type.

The circulation can be accelerated at any time by means of the double suction brass-fitted centrifugal pump driven by a belted electric motor, and installed on a by-pass just back of the boilers. This pump has a capacity of 250 gallons per minute against a 10-ft. head.

The radiating surface consists of about 7,000 sq. ft. of direct radiators, 900 sq. ft. of gravity indirect, and about 2,000 sq. ft. of primary indirect, or tempering coils, at the fan. This radiation is all supplied by the two boilers each of which has a grate 48″ x 54″ in area. The stack is of black iron, 30″ diameter by 53′ -0″ high, and is built in five separate sections butted together and held in place by 4″ bands on the outside and lugs on the inside, as described in the specifications for a steel stack in the chapter on "Chimneys for Heating Boilers."

The details of this system have been very carefully worked out and are given at length in an article by *C. L. Hubbard* in the *Engineering Review* for April, 1911.

CHAPTER XI

ELECTRICAL HEATING

THEORETICAL RELATIONS

General Principles. The use of electrical energy as a heating agent is entirely possible, due to the heating effect produced in a wire or metallic ribbon of high resistance when it is used as a conductor for either direct or alternating current. This heating effect is definite for a wire or ribbon of given material and dimensions when the potential head under which the current flows is fixed.

Heat Equivalent of Electrical Energy. The relations involved, and the underlying principles of electrical heating may therefore be readily expressed by the following equations:

$$C = \frac{E}{R} \quad \ldots \ldots \quad (1)$$

$$R = \frac{kl}{a} \quad \ldots \ldots \quad (2)$$

$$W = C \times E = C^2R \quad \ldots \ldots \quad (3)$$

$$H = 3.415\ W = 3.415\ C^2R \text{ per hour} \quad \ldots \ldots \quad (4)$$

where C = *current* flowing, in *amperes*, E = *electromotive force* or potential difference in *volts*, R = *resistance* of the conductor in *ohms*, k = *coefficient of specific resistance in ohms* per sq. in. per ft. of conductor, l = *length* of conductor in ft., a = *area* of cross-section in sq. in., W = *watts*, or volts times amperes, H = *heat effect in B.t.u.* per hour.

It is therefore evident that *the heating value of a conductor varies as the square of the current flowing and directly with the resistance*. But for a given potential the *current varies* inversely as the resistance as shown by equation (1), hence increasing the resistance only, in such a case, would actually cut down the heat supplied. The *resistance varies* with the material, with the length of conductor, and inversely as the area of the conductor or the square of its diameter, as shown by equation (2). For a conductor of given material and length, subjected to a definite voltage, there is a certain diameter and corresponding resistance which will give the maximum heating effect, which can be determined from the above equations.

The relation between power, heat, and electrical energy is given by the following equations:

1 horsepower = 746 watts.
1 horsepower min. = 33,000 ft. lb. per min.
1 B.t.u. = 778 ft. lb.

$$1 \text{ horsepower hour} = \frac{33{,}000. \times 60}{778} = 2545. \text{ B.t.u. per hour.}$$

$$1 \text{ watt hour} = \frac{2545}{746} = 3.415 \text{ B.t.u. per hour.}$$

1,000 watts = 1 kilowatt = 1 kw.
1 kw. hour = 3415 / 150 = 22.7 sq. ft. water radiation.
1 kw. hour = 3415 / 250 = 13.6 sq. ft. steam radiation.

ECONOMY AND COST OF ELECTRICAL HEATING

Governing Factors. The economy of electrical heating depends entirely on the source or the cost of generating and distributing the electricity, since all the energy supplied the heater is converted into heat. In addition to this, account must be taken of the simplicity, additional convenience, reduced attendance and repairs, and small space occupied by the electrical equipment.

Comparative Costs. The actual money value of these factors is rather hard to determine, and hence any comparison with other forms of heating is usually based on heating effect secured per pound of coal burned, or per dollar paid for fuel and electricity. It is only when electricity can be generated at a very low cost, as in certain hydro-electric plants, that it can be used for heating on a basis of equal cost compared with the ordinary steam or hot water system of heating, as the following example will readily show.

Example. Given a house heating boiler burning coal of 12,000 B.t.u. per pound with 60 per cent efficiency; then the coal required to give the heating effect of 1 kw. hour is $\frac{3415}{0.60 \times 12{,}000} = 0.47$ lb. On the other hand the cost to generate 1 kw. hour in a turbo-generating station with 90 per cent combined mechanical efficiency is 1.00/0.90 or 1.111 kw. hour for each kw. supplied for heating. Assuming the generating set has a thermal efficiency of 15 per cent, then the coal required at 60 per cent boiler efficiency is $\frac{1.111}{0.15} \times \frac{3{,}415}{0.60 \times 12{,}000} = 3.5$ lb. in order to produce 1 kw. for electrical heating. The relative cost for fuel at the same price per ton is therefore $3.5/0.47 = 7.5$ times as much when electrical generation by steam is employed.

This comparison takes no account of interest on plant, depreciation, labor costs, etc., which would increase the value of the ratio very decidedly.

A comparison based on *cost of district heating and electrical service*, at commercial rates applicable to consumers of fairly large amounts, gives a ratio of over 40 to 1 in favor of steam, as follows: Steam at 50c per 1,000 lb. and electricity at 7.5c per kw. hour will allow $1000 \times 1000/50 = 20{,}000$ B.t.u. at 1 cent for steam, and $3415/7.5 = 455$ B.t.u. at 1 cent for electricity or a relative heating effect of $20{,}000/455 = 44:1$ for the same cost in favor of steam manufactured and distributed from a central plant.

As a final comparison consider *the small steam heating plant*, with 60 per cent efficiency, burning anthracite coal at $7.00 per ton and we find that 1c. will supply $(12{,}000 \times 2000 \times 0.60)/700 = 20{,}600$ B.t.u. for heating. Electricity will cost 10 cents per kw. hour for such a plant or 1 cent will buy 341.5 B.t.u. giving a relative cost of $20{,}600/341.5 = 60:1$ for electrical heating, as compared with the cost of the fuel required to do the same amount of heating by steam. A comparison using hot water instead of steam would give the same ratios as above.

Efficiency of Electrical Heating. The efficiency of electrical heaters placed directly in the room is 100 per cent, but there is some small loss due to transmission through the wiring system of the building and feeders. When electric immersion heaters are used for heating water in tanks, the heat loss due to radiation from the tank may reduce the actual water heating efficiency to 90 or 95 per cent.

In a similar way if electric grid heaters be used for warming air for ventilation, the air heating efficiency measured in heat supplied the apartment may be only 90 to 95 per cent due to radiation losses from ducts and casings.

The comparative cost of *electricity for cooking* where the coal is necessarily burned very inefficiently may prove less than the use of coal. Under such conditions the heat usefully applied to cooking purposes may prove not over 1 per cent when coal is used, while possibly 90 per cent of the heat generated by the electrical heater is utilized in cooking.

The comparative *cost of installing* a permanent electrical heating system as compared with a hot water system to heat the same building was estimated at $1,500 for the former as against $1,860 for the latter type of installation.

ELECTRICAL HEATERS

Direct Radiators. The commercial electrical heaters of the *non-luminous* type (Figs. 1 and 2) are generally coils of special resistance wire or ribbon so proportioned as to length and cross-section as to produce a fairly low coil temperature under the voltage for which they are designed to operate. This temperature should be kept constant and the heating effect controlled by varying the number of conductors in the heater. These conductors are usually coils of wire wound spirally on an incombustible clay core, and arranged for maximum air circulation. The

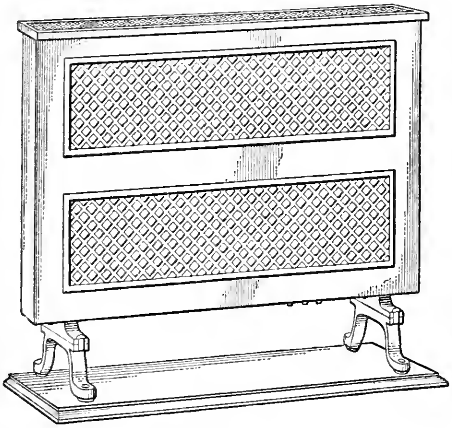


FIG. 1. SIMPLEX ELECTRIC RADIATOR.

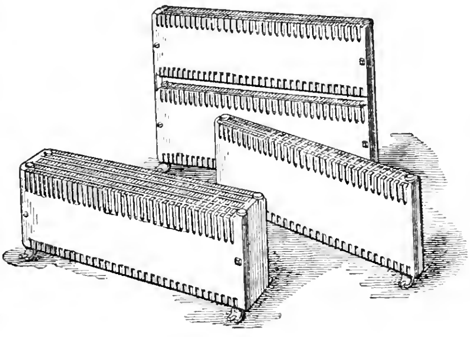


FIG. 2. ELECTRICAL CAR HEATER.

wire used is a special alloy that will not oxidize, and the coils must be placed under sufficient initial tension to prevent them from creeping out of place when under heat.

The following *capacity ratings and dimension data* apply to the non-luminous radiators made by the *Simplex Electric Heating Co.*, for permanent house heating installations. (Fig. 1.) These radiators may be regarded as practically non-portable.

TABLE 1

Simplex Electric Radiators, 2000 to 6000 Watts.
Coiled Spring Type. Volts 100–125, 220–250 and 500. Watt rating based on 112, 230, and 500 volts.

For One Heat Only

No.		Watts
104	Radiator. Base, 27 x 7¼ in. Height, 23 in. Weight, 70 lb. One heat. 8 feet cord; no plug. All voltages	2000
105	Radiator. Base, 27 x 10½ in. Height, 23 in. Weight, 95 lb. One heat. No cord. No plug. All voltages	4000
106	Radiator. Base, 27 x 14 in. Height, 23 in. Weight, 115 lb. One heat. No cord. No plug. All voltages	6000

For Three Divisions of Heat

No.		Watts Min.	Watts Max.
101	Radiator. Base, 27 x 7¼ in. Height, 23 in. Weight, 70 lb. Three heat. Connector switch. No cord. No plug. Not made for 500 volts	670	2000
102	Radiator. Base, 27 x 10½ in. Height, 23 in. Weight, 95 lb. Three heat. Connector switch. No cord. No plug. All voltages	1335	4000
103	Radiator. Base, 27 x 14 in. Height, 23 in. Weight, 115 lb. Three heat. Connector switch. No cord. No plug. All voltages	2000	6000

Luminous radiators (Fig. 3) are very desirable for localized and temporary heating since their radiant energy is immediately available, and it is not necessary to wait until warm air currents have been established before their heating effect is felt. These radiators or glowers

are simply large incandescent lamps of low candle-power but high watt capacity. Up to 500 watts they may be used on ordinary lighting-circuits, but above this capacity a special heating circuit is necessary. These luminous radiators are always portable, and therefore one outfit may serve several rooms.

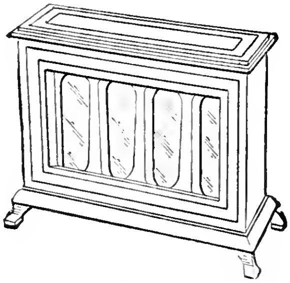


FIG. 3. FOUR-GLOWER ELECTRIC RADIATOR.

Length of Coil Required. Since the heating effect, $H = 3.415\ C\ E$ in B.t.u., and $R = k\,l\,/\,a$, the length of coil (l) in ft. for a given size or area (a) in sq. in. of wire is readily found by substitution, using equations (1) to (4).

$$H = 3.415 \times C \times E = 3.415\frac{E^2}{R},$$

and by equation (4)

$$H = 3.415 \times \frac{E^2 \times a}{k \times l},$$

from which $l = \dfrac{3.415}{H} \times \dfrac{E^2 \times a}{k}$ = length of coil in feet for the given voltage (E). The specific resistance (k) depends on the material of the wire or ribbon and must be stated in ohms per sq. in. per ft. of length, the value of which should be taken from a table of electrical conductivities. Allowance should be made for an increase in the resistance of the wire due to its rise in temperature otherwise the expected heating effect will not be obtained unless a wire with zero temperature coefficient is used.

The *current consumption and size of heating unit* in ordinary rooms at 70° F is estimated at from ¾ to 1½ watts per cu. ft. per hour with 0° outside and non-luminous radiators. Hence for a room 12 x 12 x 10 = 1440 cu. ft. the minimum heating requirements would be ¾ × 1440 = 1080 watts per hour. This is only a rough approximation and should not be resorted to unless the actual heat loss in B.t.u. cannot be calculated. If H represents this loss in B.t.u. per hr. then $H/3.415 = W$ or watts per hour to be given off by the radiator. The heat loss should be calculated as already given in the chapter on "Heat Transmission of Buildings."

Car Heaters. In addition to the commercial heaters designed for house or office service (Figs. 1 and 3) there are those designed for car heating (Fig. 2) where special conditions must be provided for in the way of very severe space limitations.

Car heaters are usually connected in series, six heaters to the average car, requiring about 8 amperes at 500 volts or 4,000 watts per car. This is equivalent to 5½ horsepower for heating as compared with 15 horsepower for average running conditions. Records from 155 street cars, 38′-8½″ long x 7′-10″ wide x 8′-6⅞″ center height and 6′-7¾″ side height, with 208 sq. ft. of glass surface per car, show that at least 7,000 watts per car are required for heating cars of this size.

It is very common practice to allow 1 electrical horsepower for heating (by resistance) as the equivalent of 7.5 sq. ft. of ordinary steam radiation at 135° above the room temperature of 70°. This is equivalent to allowing 100 watts as equal to 1 sq. ft. of steam radiation in heating effect and is quite safe.

INSTALLATION OF ELECTRICAL HEATERS

The installation of complete electrical heating systems is confined almost entirely to electric railway cars, and very few permanent installations have been made in buildings, unless rates for electric current or power were phenomenally low. There have been, however, a few complete plants installed and in one case a schoolhouse has been heated by a fan system using both direct and indirect electrical radiators.

TABLE 2

SUGGESTIONS FOR A HEATING AND POWER CIRCUIT IN A FOUR-STORY CITY RESIDENCE HAVING BASEMENT AND SUB-BASEMENT

The circuit to be 220 volts, connected to a separate power meter

	Room	Device	Outlet Ampere Capacity	Total Ampere Capacity
Sub-basement	Cellar	Ice machine	3.5	
		Passenger elevator	25	
		Ash-lift	3	
		Vacuum cleaner	10	41.5
Basement	Kitchen	Instantaneous water heater	20	
		Kitchenette	30	
		Exhaust fan	4	
	Butler's room	10-lb. pressing iron	3.5	
	Laundry	Clothes dryer	20	
		Wash boiler	10	
		4-lb. iron	3	
		7-5-lb. iron	3	93.5
Main floor	Dining-room	Chafing dish, percolator, and so forth (outlet placed in floor under table)	3	
		Portable radiator (outlet placed in wall)	10	
	Music-room	Piano player	1	
	Parlor	Portable radiator	10	
	Pantry	Plate warmer	5	
		Polishing motor	1	30.0
Second floor	Library	Radiator in fireplace	10	
	Bedroom	Portable radiator	10	
	Front bedroom	Hair dryer, iron, sewing machine, and so forth	10	
	Bath-room	Water cup, hair dryer	2	
		Portable radiator	10	42.0
Third floor	Bath-room	Radiator, vibrator, and so forth	10	
	Front bedroom	Pressing iron, and so forth	3	
	Middle bedroom	As above	3	
	Rear bedroom	Water cup, heating pad, and so forth	3	19.0
Fourth floor	Play-room	Portable radiator	10	
		Electrical toys	2	12.0
		Total		238.0

An installation similar to the above would cost about $850, exclusive of the devices. This figure is based upon a new house and the circuit independent from the lighting circuit. The irons might be placed on the 110-volt lighting circuit, which would reduce the cost of the above installation.

The *wiring system* must be so designed that there will be no appreciable drop in voltage or no heating effect in the mains and radiator connections. The drop in voltage must be confined

to the heaters, which are connected on a two-wire system properly fused and provided with main line and individual radiator switches of proper capacity for controlling the system.

All wiring should be run in suitable conduit in accordance with the specifications of the *National Board of Underwriters*, and, if desired, a system of automatic temperature control may be applied to the electrical radiators to maintain constant temperature conditions in the heated rooms.

The necessity for providing ample carrying capacity in all heating circuits usually requires that such apparatus be wired separately from the lighting service, and possibly at a higher voltage, say 220 volts instead of 110 volts as for lights. The foregoing table by *C. D. Wood, Jr.*, of the *National Electric Light Ass'n*, will give an idea of the heating and power requirements to be met by such service in a large residence.

CHAPTER XII

VENTILATION, AIR ANALYSIS AND VENTILATION LAWS

VENTILATION

Ventilation. Ventilation, whether *natural* or *mechanical*, consists in the displacement of vitiated air from an apartment and its replacement by fresh air. To state that the air in an apartment is renewed any given number of times per hour is not strictly accurate, as a positive change does not actually occur; the incoming air mixes with and dilutes the foul air to a point suitable for healthful respiration.

In *natural ventilation* systems the movement of the air in flues, ducts, etc., is induced solely by the thermal head produced by the *difference in density* between the column of air in the ducts and that of the outside atmosphere; the higher the temperature in the ducts the more rapid does the draft become.

The direction and velocity of the wind outside of a building will materially affect the natural ventilation, retarding or accelerating the movement of the air through ducts and flues, according to the exposure of the building and the position of inlets and outlets in relation to same.

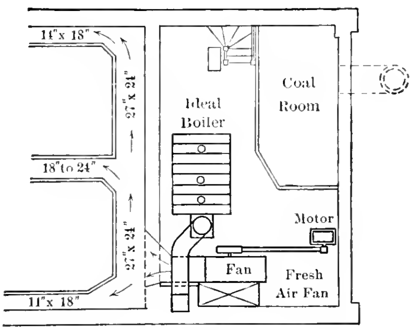


FIG 1. UPWARD VENTILATION SYSTEM.

In *mechanical ventilation* the movement of air is maintained by means of various types of fans, either driven by a steam engine, electric motor, or other prime mover. With fans of known efficiencies the results can be accurately estimated, as shown in the chapter on "Hot Blast Heating."

The principal advantages of the use of mechanical systems of heating and ventilation are:

1. Positive circulation of air at any desired temperature to all parts of the building

2. Practical uniformity of ventilation regardless of indoor and outdoor temperatures and atmospheric conditions, as a positive movement of the air is possible through all ducts, flues, etc., at any required pressure.

Systems of Ventilation. *Ventilation systems* are also broadly divided into two general classes known as the upward and downward systems.

The former or *upward system* (Fig. 1) is generally used for audience rooms where there is a

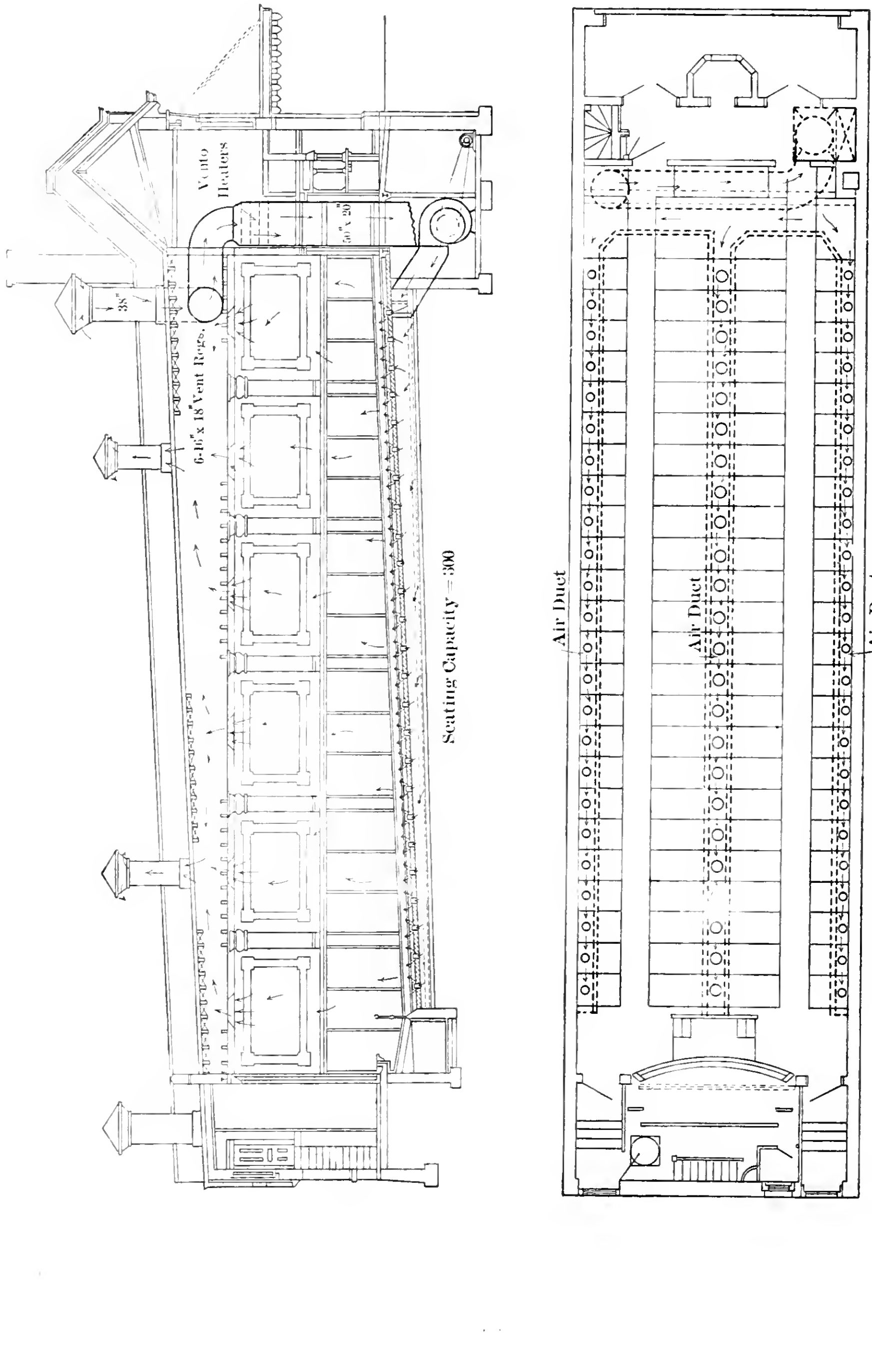


FIG. 1. (*Continued.*) UPWARD VENTILATION SYSTEM.

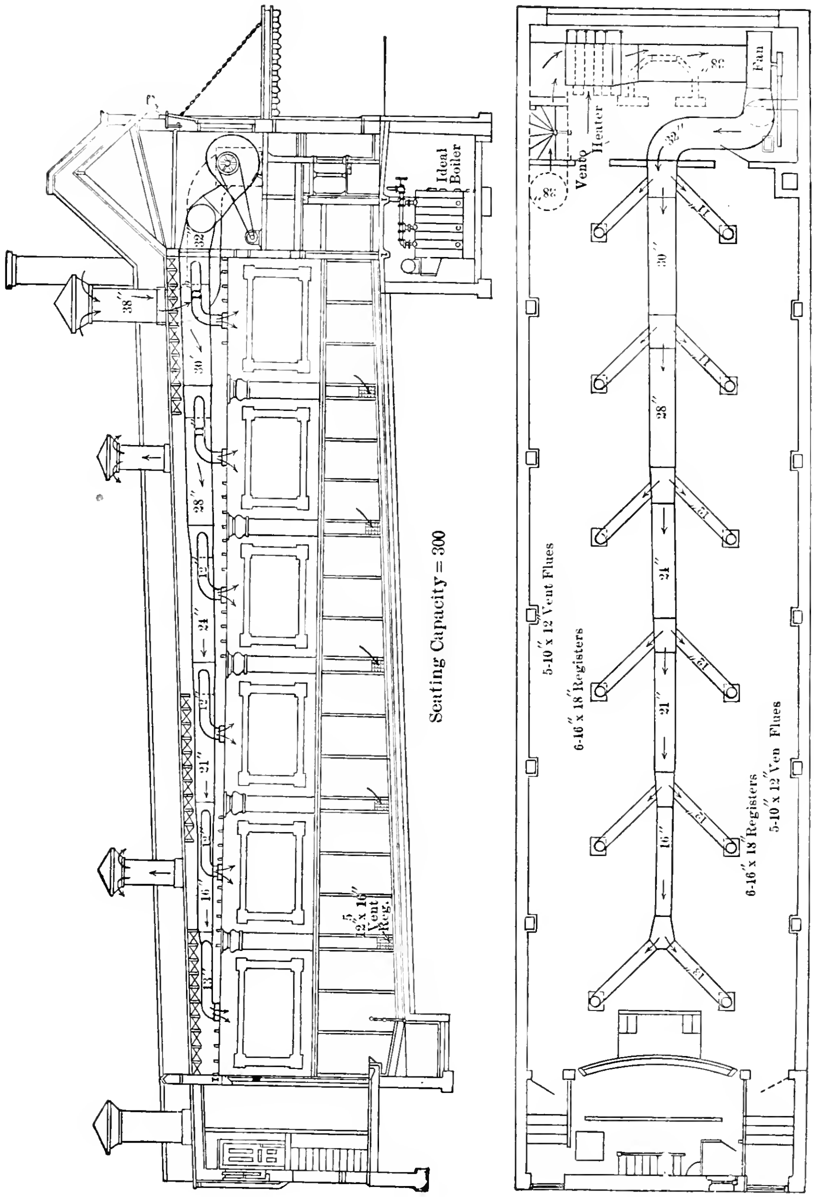


FIG. 2. DOWNWARD VENTILATION SYSTEM.

strong natural tendency for the heat given off by the large number of occupants to rise and take with it the vitiation products due to respiration. The *air is supplied near the floor-line* through mushroom ventilators in the floor, or through the hollow pedestals of the chairs themselves, or low registers. The *vitiated air outlets are in or near the ceiling.* This system makes it rather difficult to heat the room in advance of the arrival of the audience as the outlets allow the warmed air to escape almost as rapidly as it can be introduced.

The latter or *downward system* (Fig. 2) is very generally used in school-rooms, hospitals, institutions, etc. The occupants are not as closely placed as in the former case, and a more even distribution of air and more uniform heating can be secured when the *air is supplied 8 feet or more above the floor,* and the *vitiated air removed at or near the floor-line.* Due to the elevation of the inlets above the heads of the occupants there is little liability of drafts, and if the outlets are on the same side-wall as the inlets there is very little opportunity for short-circuiting between inlet and outlet, since the incoming air must flow out across the room to the cold outside wall before it can cool and drop to the floor-level. It is, however, necessary in the downward system to overcome the natural tendency of the heated air from the bodies of the occupants to rise, and oppose the uniform downward tendency of the incoming fresh air.

It is absurd to attempt to characterize one system as superior to the other, as the selection of either system must depend entirely on the conditions to be met, which have been outlined in the above paragraphs.

Distribution of the Air. In general, it should be observed that whether upward or downward ventilation is employed there should always be a definite system of vitiated air removal, designed to provide for uniform distribution and prevent short-circuiting between inlets and outlets. The diffusion efficiency of various arrangements of air-inlets and outlets is shown in Fig. 3—*a, b, c, d* and *e*. The tendency for the incoming fresh air, especially if heated, to short-circuit from inlet to outlet is shown in conditions *a, b, c* and *d*. In all of these cases the tendency of the fresh air to pass through the room above the breathing line is very apparent.

The practically complete diffusion shown in condition *e* can only be attained when inlet and outlet are placed in the same inside wall, with the former at least 7 to 8 ft. above the latter.

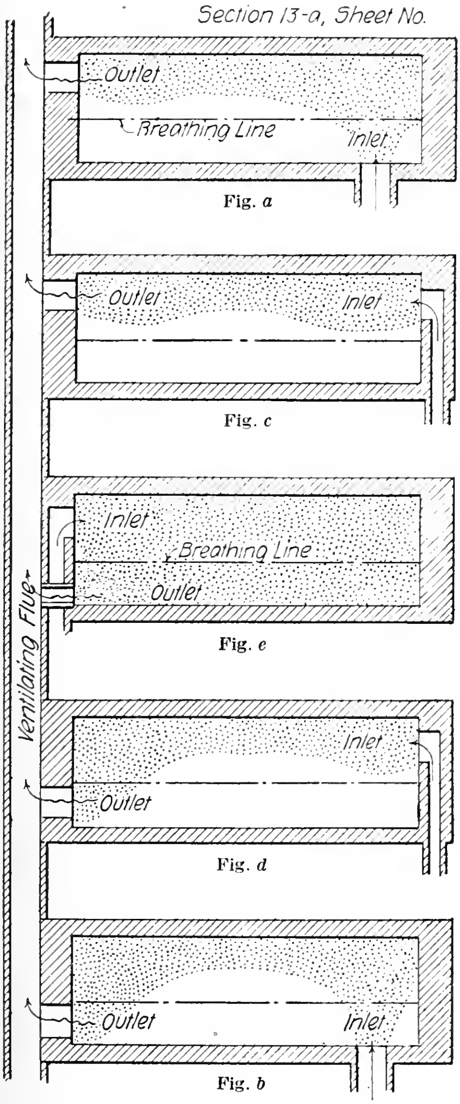
The use of *multiple inlets* and *mushroom ventilators,* in order to secure a better mechanical distribution of the air, is being practised in many systems of upward ventilation for audience rooms with fixed seats.

In this case a false floor or plenum chamber must be constructed just below the main floor through which the air is to be supplied.

Mushroom ventilator heads (Figs. 4 and 5) are then located under every second or third seat and adjusted to give a uniform discharge of tempered air over the entire seating area. These heads are either mounted on an adjustable spindle (Fig. 4) which is supported centrally in the cast-iron floor sleeve or flange or else have a non-adjustable spindle (Fig. 5) similarly supported, and are equipped with a control damper. In either case the adjustable head or damper must be locked positively in the finally adjusted position.

In the case of concrete floors it is very desirable to use a cast-iron sleeve and flange (Fig. 4), rather than a galvanized sleeve and cast-iron flange (Fig. 5). The former is made by the *American Blower Co.* and the latter by the *Knowles Mushroom Ventilator Co.* The last named company also make a cast-iron head and sleeve with adjustable head. In any case the sleeve of thimble should be cast or made just the thickness of the floor.

Necessity for Ventilation. The necessity for ventilation depends on the use to which the building or rooms are to be put. Wherever human beings are present in closed quarters *the gaseous products of respiration* should be removed as rapidly as possible, and fresh air supplied. Likewise in those cases where *the gaseous products of combustion* from artificial illumination or chemical processes are given off in the room they should also be removed. The *heat generated by the occupants, lights, machinery and in other ways* may prove excessive, and the ventilation system should provide for its removal or dissipation. Finally, it may be that *the moisture (water vapor) given off by occupants, lights, or processes* will increase the relative humidity to such



Effect of Locations of Inlets and Outlets on Air Distribution in a Room.

FIG. 3.

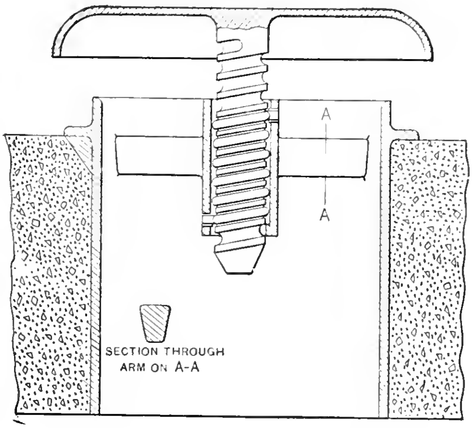


FIG. 4. SECTION THROUGH "ABC" MUSHROOM VENTILATOR.

SIZES OF "ABC" MUSHROOM VENTILATOR.

Size.	Approximate Inside Diameter.	Approximate Weight.
4	4¼″	6 lbs.
5	5½″	10 "
6	6¾″	15 "

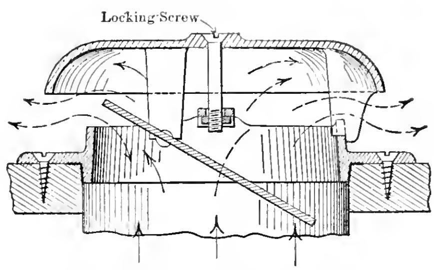


FIG. 5. KNOWLES AIR CONTROLLING HEAD AND DIFFUSER—SINGLE DAMPER STYLE.

Sizes of Knowles Air Controlling Head and Diffuser, Single Damper Style. Made of cast iron.

4⅝-inch opening........6-inch head
5 " " ....6 " "
8 " " ....9 " "

Special sizes will be made if desired.

an extent as to cause discomfort to occupants or interfere with the said processes, due to condensation, and the ventilation system must provide for removing same. Again it may happen, and in general this is more usual, that *insufficient moisture* will be present in the air, and this deficiency must also be supplied by the ventilation system.

Any or all of the above reasons may require the installation of a system of ventilation for supplying fresh air, suitably conditioned to overcome or ameliorate the conditions that would otherwise exist.

The vital importance of supplying buildings with fresh air is each day becoming more generally recognized. That the ill effects of repeated and protracted exposure to foul air results in the slow but sure production of throat and lung trouble has been conclusively shown by the repeated observations of many eminent physicians, both in this country and abroad.

The reduction of the death rate as shown by records covering a considerable number of years in prisons and hospitals, that has resulted when proper provision for ventilation has been inaugurated, has been in many instances remarkable. In this connection *Prof. S. H. Woodbridge* states: "Prison records show reduced death rates chiefly as a result of effective ventilation. In one case the rate diminished from a yearly average of eighty deaths to one of eight, each period covering the same and a considerable number of years."

W. G. Snow states in his paper, "Ventilation in its Relation to Health": "In certain buildings where the results of changing from poor to good ventilation have been carefully observed, a marked improvement in the general health of the occupants has been manifest. For example, the records of the *United States Pension Bureau* show that when the offices of the department were located in scattered and poorly ventilated buildings, 18,736 days were lost by employees through illness in one year and about the same number for several successive years. When the department became established in its new, well ventilated quarters, the loss was reduced to 10,114 days on account of illness, the working force being larger and the work increased. The gain effected is not to be measured alone by the days' absence saved, but by the greater vitality and efficiency of the entire working force."

The Effect of Vitiated Air. The amount of carbon dioxide present in vitiated air has been until recently quite generally understood to be the element of danger that should be kept within safe limits.

Dr. Remsen has pointed out that the presence of carbon dioxide in itself is not dangerous to health except in that it reduces the supply of oxygen by displacing it.

Carbon dioxide is not poisonous, but the organic impurities that are exhaled at the same time with other gases that are given off may prove a menace to health. The ill effects of breathing air in a poorly ventilated room are due to the small quantities of decomposing organic matter and unhealthy gases. The carbon dioxide that has been generated by the lungs and is given off at the same time serves more or less as an indicator of the presence of the real danger. Any lowering of the oxygen supply that is actually required for the proper and necessary transformation of the potential heat value of the food into the physical and nervous energy required to keep the human machine running, and to readily supply the additional demand made upon that machine to perform external work, means that industrial workers who perform their duties in a vitiated atmosphere do so at the expense of a lowered vitality, and are naturally less productive.

The Object of Ventilation. Satisfactory ventilation not only consists in constantly supplying fresh air, but in furnishing this air in a pure condition, *free from dust and other impurities,* at the proper temperature and *with the proper amount of moisture present,* but also in the efficient removal of the vitiated air. This cannot be positively and satisfactorily accomplished during the heating season by the simple expedient of opening doors and windows unless some mechanical means is employed.

Relation between Humidity and Temperature. The proper and healthful relative humidity of the air in buildings has only in recent years been given the thought and attention it right-

fully deserves. Heated or warmed air, whether it be that which may be purposely introduced into a building for warming, or which naturally enters by infiltration, on being expanded by heat will have its percentage of moisture or relative humidity lowered with the consequence that the capacity of the air for absorbing moisture has been greatly increased. We consequently experience the sensation of so called *dry heat*. This causes an excessive and unnatural evaporation of moisture to take place from the skin and membranes of the respiratory organs.

Evaporation takes place by the direct application of heat and is essentially a refrigerating or cooling process. Heat being abstracted from the body for this purpose, naturally tends to lower the surface temperature, and we actually feel several degrees cooler than the temperature recorded by the thermometer in the room.

The following chart (Fig. 6) may be used to show the relation between the temperature and relative humidity of rooms, and their bearing on the comfort of the occupants. This chart establishes a so-called "Comfort Zone" in which the relation between relative humidity and temperature is given by the equation, $R = 100 - 4\ (T - 54)$, the value of which is shown by the dotted line plotted on the sheet.

In this connection *Dr. Henry Mitchell Smith*, in a paper read before the *Brooklyn Medical Society*, says, in effect, that the records from a large number of observations show that the tem-

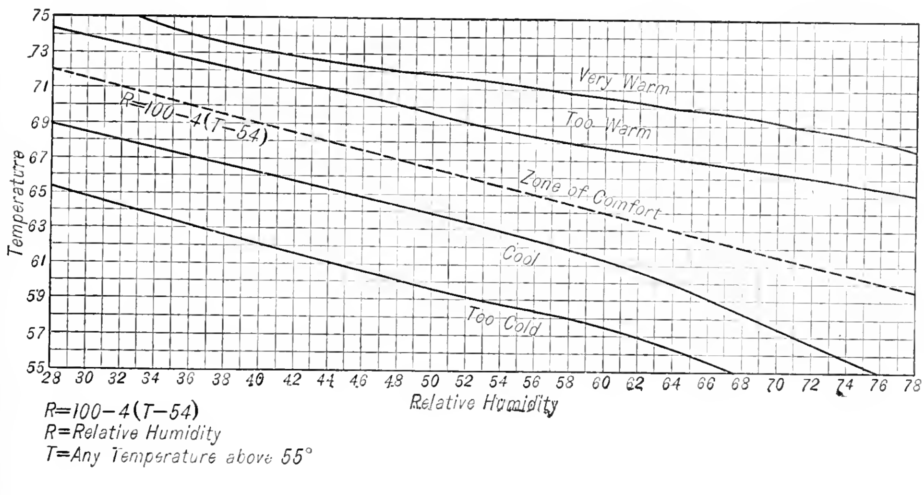


FIG. 6. RELATION BETWEEN HUMIDITY AND TEMPERATURE.*

perature in city residences commonly range from 72° to 76° F. in cold weather with a relative humidity of about 30 per cent. Rooms under these conditions often felt chilly and uncomfortable to the occupants.

Dr. Smith's many observations and experiments upon the sensations produced by different percentages of saturation, led him to make the following statement: "It may be accepted as a cardinal rule that if a room at 68° is not warm enough for any healthy person, it is because the relative humidity is too low."

* NOTE.—The above chart is based on the results of "Ventilation Tests at The Chicago Normal College" by the *Chicago Commission on Ventilation*, under the direction of *Dr. E. Vernon Hill*. The data in the complete report show that a temperature of 64° F. with 55% relative humidity was as "comfortable" as a temperature of 70° F. with only 30% relative humidity.

A room in which the temperature is kept at approximately 68° with a relative humidity of 50 per cent, will feel warmer than a room having a temperature of from 72° to 74° with a relative humidity of 25 to 30 per cent. *Dr. Smith* further says: "The point to be emphasized is that every time we step out of our houses during the winter season we pass from an atmosphere with a relative humidity of about 30 per cent into one with a relative humidity of an average of 70 per cent. Such a sharp and violent contrast must be productive of harm, particularly to the delicate mucous membranes of the upper air passages.

"The skin and the mucous membranes of the respiratory passages are the principal sufferers, since these tissues are always kept moist with their own secretions; and from them water is freely abstracted to satisfy this large saturation deficit.

"A moment's consideration shows that the prevailing practice of depending upon the thermometer as the sole guide in the heating of buildings is not only inadequate and unscientific, but it is misleading. It is *not sufficient to know only the temperature* if we desire either comfort or health, for the same temperatures produce varying sensations of warmth or chilliness, depending upon the relative humidity at the time existing.

"It is unscientific and arbitrary to lay down a fixed temperature as a standard for living- or sleeping-rooms unless the relative humidity is indicated as well."

Economy of Ventilation. From an economic standpoint alone, the modern manufacturer cannot afford to overlook the direct bearing that *good ventilation* has upon the cost of production, and to consider the heating of a shop or factory without at the same time investigating the question of proper and adequate ventilating facilities.

"The Cash Value of Factory Ventilation" is the subject of an interesting article by *Professor Winslow*, in which he makes the following statement: "Efficient production requires skilled and practiced workers, in good physical condition, applying themselves with energy and enthusiasm to their tasks. Irregularity of attendance and the physical sluggishness and nervous inattention which accompany lowered vitality mean direct money loss to the employer of labor, as well as a burden on the community at large."

Compulsory ventilation requirements are in force in a few states and are practically due to the result of the efforts of members of the *American Society of Heating and Ventilating Engineers*. This Society is constantly at work to secure the passage of similar laws in other states.

The laws thus far passed apply principally to the ventilation of school buildings. Some states, however, have passed factory ventilation laws, although as a rule these are not by any means rigorously enforced.

A list of the states having compulsory ventilation laws will be found at the end of this section and also a specimen law.

Causes of Air Vitiation. The vitiation of the air in a building as understood with respect to ventilation may be due to (1) *an excess* of *carbon dioxide* due to respiration, lights, etc., above that found in outside air, (2) *the generation of heat* by the occupants, lights or other causes above that required for warming, (3) *an excess or deficiency of water vapor* above or below that required for the desired relative humidity, (4) *the presence of dust* either brought in by the air or created by the operations within the building, (5) *the presence of bacteria* due to respiration or otherwise and (6) *the presence of odors* due to occupants or manufacturing processes.

It will be evident from the above that the ventilation of a building must provide for every condition of the air within same except warming the air above the temperature at which the building is to be maintained.

The following data relative to *respiration, illumination, manufacturing processes,* etc., is taken from standard authorities.

Effect of Respiration. The effect of respiration is shown by Table 1.

TABLE 1

COMPOSITION OF PURE AIR AND RESPIRED AIR

(*R. C. Carpenter*)

Constituent	Pure Air		Respired Air
Oxygen	20.26%		16.2%
Nitrogen	78.00		75.0
Carbon dioxide	.04		4.0
Water vapor	1.50	Variable	5.0

This expired air leaves the lungs at a temperature of 90° to 98° F., is nearly saturated with water vapor, and is from 1 to 3 per cent lighter than when inhaled.

It will be seen that the greatest change has occurred in the per cent of CO_2 which has *increased about 100 times*, or from 4 parts in 10,000 to 400 parts in 10,000, as exhaled from the lungs.

Diffusion of the exhaled gases takes place at once, and the CO_2 is soon distributed quite uniformly throughout the air of the room, thereby increasing the CO_2 content of the room air above 4 parts in 10,000. This phenomenon of diffusion requires time, the interval varying inversely as the density and directly as the square root of the absolute temperature of the gas.

When the CO_2 content due to respiration exceeds 7 parts, the effect of poor ventilation is easily apparent, and at 10 parts in 10,000 actual discomfort and headache are experienced. It is therefore customary to *limit the increase* in CO_2 in parts per 10,000 above that in outside air to from 2 to 3 in a well-designed system of ventilation, and the cu. ft. of outside air to be supplied for each person in order to maintain such a standard is easily calculated as follows:

An adult at rest requires about 20 cu. in. of air at each respiration and will make from 16 to 24 respirations per minute, so that a total of from 320 to 480 cu. in. of air are required per min., or about 0.25 of a cu. ft. This amount may be increased by exercise to 0.33 of a cu. ft. Since this exhaled air contains about 400 parts CO_2 per 10,000, an adult at rest gives off $0.25 \times 60 \times 0.04 = 0.6$ cu. ft. of CO_2 per hr. Some authorities allow 500 cu. in. of air and 17 cu. in. of CO_2 per minute. Hence we must supply for each adult in order to maintain a CO_2 content of 7 parts with outside air containing 4 parts per 10,000,

$$C = \frac{a \times b}{n - 4} = \text{cu. ft. per min.,}$$

where a and n = parts of CO_2 per 10,000 for respired air and standard of purity respectively, and b = cu. ft. per min. = 0.25 ordinarily.

Then $C = \dfrac{400 \times 0.25}{7 - 4} = 33.3$ cu. ft. per min. per person or equivalent, or 2,000 cu. ft. per hour for a standard of 7 parts in the room, and 4 parts in 10,000 in outside air. The latter value is practically constant and Tables 2 and 3 have been calculated on this basis.

TABLE 2

VALUES OF C FOR VARYING VALUES OF n FROM 5 TO 20

Vitiation Allowed, Parts of CO_2 per 10,000 in the Room Air	Cu. Ft. of Air Containing 4 Parts of CO_2 per 10,000 to be Supplied per Person or Equivalent. A Person = 0.6 Cu. Ft. CO_2 per Hr.	
Parts = n	Per Minute = C	Per Hour
5.0	100.0	6000
5.5	66.6	4000
6.0	50.0	3000
6.5	40.0	2400
7.0	33.3	2000
7.33	30.0	1800
7.5	28.6	1714
8.0	25.0	1500
9.0	20.0	1200
10.0	16.6	1000
15.0	9.1	545
20.0	6.2	375
30.0	3.8	231

TABLE 3

VOLUME OF AIR NECESSARY FOR GIVEN STANDARD OF PURITY

Cubic Feet of Space in Room per Individual	Proportion of Carbonic Acid in 10,000 Parts of the Air Not to be Exceeded at End of Hour						
	6	7	8	9	10	15	20
	Cu. Ft. of Air, of Composition 4 Parts of Carbonic Acid in 10,000, to be Supplied the First Hr.						
100	2900	1900	1400	1100	900	445	275
200	2800	1800	1300	1000	800	345	175
300	2700	1700	1200	900	700	245	75
400	2600	1600	1100	800	600	145	None
500	2500	1500	1000	700	500	45	...
600	2400	1400	900	600	400	None	...
700	2300	1300	800	500	300	...	...
800	2200	1200	700	400	200	...	...
900	2100	1100	600	300	100	...	...
1000	2000	1000	500	200	None	...	...
1500	1500	500	None	None	...	...	...
2000	1000	None	....	....	...	...	...
2500	500	....	....	....	...	...	...

The amount of air supplied per person for various classes of service in the best ventilation practice is shown in Table 4.

TABLE 4

AIR REQUIRED FOR VENTILATION

Type of Building		Cu. Ft. Air Required per Hr. per Person
Hospitals	Ordinary	2500
	Surgical cases	3000
	Contagious diseases	6000
Schools, theaters, prison and assembly halls		1800
Factories and shops		2000
Factories (unhealthy trades)		3500

In some cases the amount of air to be supplied is based on the number of air changes which will occur due to natural infiltration, or which may be mechanically supplied per hour, for which Tables 5 and 6 will serve as a guide.

TABLE 5

AIR CHANGES DUE TO NATURAL INFILTRATION WITH DIRECT HEATING SYSTEMS

(R. C. Carpenter)

Type of Building	Number of Changes per Hour
Residence	Hall, 3—Sitting-room, etc., 2—Sleeping room, 1
Stores	First floor, 2 to 3—Second floor 1½ to 2
Offices	First floor, 2 to 2½—Second floor 1½ to 2
Churches and public assembly rooms	From ¾ to 2
Large rooms with small exposure	From ½ to 1

A paper, by *John D. Small*, was presented at the Semi-Annual Meeting, 1912, *A. S. H. & V. E.* which dealt with the present office practice in allowing for air change in various types of buildings. The following table is taken from this paper and shows not only the air change due to leakage but also gives changes due to ventilation in many instances:

TABLE 6

AIR CHANGES TO BE USED AS A BASIS FOR HEATING AND VENTILATING CALCULATIONS IN VARIOUS TYPES OF BUILDINGS BOTH DIRECT AND INDIRECT SYSTEMS

(*J. D. Small*)

Type of Building	Air changes to be allowed
Office Buildings	Portions above grade— 1 change per hr.
	Basement, general — 4 change per hr.
	Mechanical plant —10 change per hr.

Factory Buildings which have no mechanical or natural ventilation, 1 change per hour. For factories where large doors from the outside are frequently opened, about 4 air changes per hour.

Residences—having loose windows, 2 changes per hour.

Churches—Four changes per hour except small rooms, which should have 5 or 6 changes per hour. These data for churches contemplate mechanical ventilation. The majority of public buildings and many of the factories require ventilation or the fan system of heating.

THE USUAL SPECIFICATIONS FOR AIR SUPPLIED PER PERSON ARE AS FOLLOWS:

Hospitals:
- Ordinary—35 to 40 cu. ft. per min.
- Epidemic—80 cu. ft. per min.
- Tuberculosis

Tuberculosis	Air Change
Detention Room	6 Min.
Toilet Rooms	6 "
Bath and Duty Rooms	8 "
Kitchen	3 "
Serving	10 "
Fumigating	10 "

Workshops—25 cu. ft. per min.
Prisons—30 cu. ft. per min.
Theaters—20 to 30 cu. ft. per min.
Meeting Halls—20 cu. ft. per min.
Schools—30 cu. ft. per min. per child and 40 cu. ft per min. per adult.

FOLLOWING TIME-INTERVALS FOR ONE AIR CHANGE ARE USUAL:

HOTELS

Room	Air Change	Room	Air Change
Engine	6 Min.	Cafe	8 Min.
Kitchen	1½–5 "	Lobby under Balcony	8 "
Restaurant	6 "	Main Lobby	20 "
Base Toilet	5 "	Banquet Hall	15 "
Billiard	10 "	Retiring Room	10 "
Barber Shop	8 "	Kitchens	8 "
Dining-Room	15 "	All others	15 "
Palm Room	12 "	Except Toilets	6 "
Buffet	8 "		

LIBRARIES

Corridors	15 Min.	Inside Rooms	8 Min.
Basement Rooms	15 "	Corner Rooms	7 "
Reading Rooms	12 "	Toilet Rooms	5 "

Laundries—should have an air change every 4 to 6 min.

NOTE.—Radiation on sides of buildings subjected to prevailing and cold winds should be increased 10 per cent up to 10th floor and 15 per cent above.

Heat and Water Vapor Due to Occupants. Respiration and the presence of occupants in a room not only increases the CO_2 content of the air but may add materially to *the heat and moisture content* of the air as shown in Table 7.

Heat in its relation to life. All life processes are accompanied by the generation of heat from within. The blood serves to carry oxygen from the lungs to remote tissues, where combustion takes place, and CO_2 is carried back and eliminated by the lungs in respiration.

As a result an average normal temperature of 98.6 °F. is maintained, and a constant but not too rapid radiation of heat must be kept up if the individual is to remain comfortable.

Hence it appears that the only *function of external warming is to reduce the rapdity with which the body parts with or gives off its heat.* An average adult must dissipate about 400 B.t.u. per hour for comfort, of which about 30 per cent is lost by contact with the air, 43 per cent by radiation and 27 per cent by exhalation and other means.

Air, if dry, is a nearly perfect non-conductor but allows radiation to take place through it readily, hence in a room with very dry air at 75° a person may feel cold if walls are at 50°, due to the radiation loss from the body to the cold wall surfaces.

It is desirable to maintain a normal rate of heat loss if comfort is to be had, and surround-

ing air and wall temperatures ranging from 67°–72° F., depending on quietness of air, are found desirable.

TABLE 7

HEAT AND WATER VAPOR GIVEN OFF BY PEOPLE

Occupant	B.t.u. per Hour	Water Vapor per Hour, Lbs.
Man at work	794	0.267
Man at rest	397	.155
Youth	357	.088
Infant	103	.029

The general effect of respiration on the air of inhabited rooms is given in the following extract from an article by *Chas. R. Honibal, Engineering Review:*

"The accompanying tables, obtained from results of experiments, represent the average of a very large number of observations on the air of inhabited rooms. The method employed in judging the quality of the air was to enter directly from the open air into the room in which the air was to be judged, after having been at least 15 minutes in the open air. It will be seen how closely the state of the room, as detected by the sense of smell, agrees with that which would be expected from the carbonic acid as shown by analysis and the hygrometric excess. In these experiments 0.0002 cubic foot of carbonic acid per cubic foot of air was taken as the standard of impurity, in addition to 0.0004 carbonic acid per cu. ft. as the normal amount of CO_2 in the outer air."

TABLE 8

Sense of Smell	Temperature Deg. Fahr.		Vapor Grains Per Cu. Ft.		Carbon Dioxide per 10,000 Vols.	
	In Air Space	Excess Over Outer Air	In Air Space	Excess Over Outer Air	In Room	Excess Over Outer Air
1. Fresh	62.85	11.38	4.629	0.344	5.999	1.830
2. A little smell	62.85	8.00	4.823	0.687	8.004	3.894
3. Close or disagreeable smell	64.67	12.91	4.909	1.072	10.027	6.322
4. Very close or offensive and oppressive smell	65.15	12.87	5.078	1.409	12.335	8.432
5. Extremely close, when sense of smell can no longer differentiate	65.05	13.19	5.194	1.319	12.818	8.817

Sense of Smell	Outer Air (W)		Room Air (W)		Purity (S)	
	Temp. Fahr.	Rel. Hum.	Temp. Fahr.	Rel. Hum.	S	Vitiation per cent
Condition 1	51.47°	80%	62.85°	72.5%	0.926	7.4%
Condition 2	54.85	87	62.85	75.6	0.856	14.4
Condition 3	51.76	87.3	64.67	73.0	0.774	22.6
Condition 4	52.28	84.73	65.15	72.7	0.772	27.8
Condition 5	51.86	88.4	65.05	76.6	0.746	25.4

Effect of Illuminants. The effect of illumination on the CO_2 content of the air is given in Table 9 which is based on the consumption of illuminant and vitiation in cu. ft. of CO_2 for each normal candle-power. It should be noted that all unenclosed lights discharge not only heat but also carbon dioxide gas, water vapor and traces of sulphur dioxide directly into the air of the room.

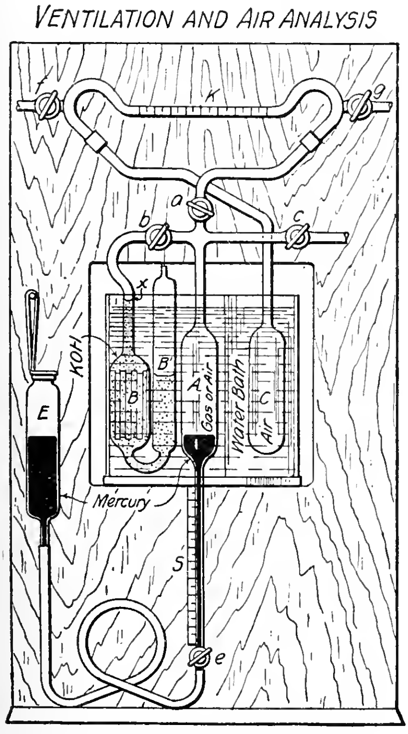
TABLE 9
VITIATION BY ILLUMINANTS
(*J. R. Allen*)

Source	Consumption of Combustible per C. P. In. Cu. Ft. per Hour	Carbon Dioxide Given off per C. P. In Cu. Ft. per Hour
Gas-Fishtail*	0.802–0.527	0.494–0.304
Gas-Argand	–0.445	0.254
Gas-Welsbach	0.053–0.024	0.030–0.057
Petroleum-Round	Gals. 0.00050	0.112
Petroleum-Flat	Gals. 0.00198	0.335
Wax candles	Oz. 0.271	0.417
Paraffine candles	Oz. 0.324	0.459

* It is readily apparent that a 16-candle-power gas flame burning gas at any rate, as for example, 5 cu. ft. per hr. will give off $(0.304/0.527) \times 5 = 2.8$ cu. ft. of CO_2 per hour which is equivalent to the vitiating effect of 4 to 5 adults. One adult produces about 0.6 of a cu. ft. of CO_2 per hr.

The *Green Fuel Economizer Co.* recommends : "For each cu. ft. of lighting gas burned per hour add 0.08 lb. carbon dioxide, 0.06 lb. water vapor and 565 B.t.u. The amount of CO_2 should not be allowed to reach 0.10 or at the most 0.15 per cent."

Atmospheric Conditions Which Fail to Support Combustion. Experiments made at the *Pittsburgh Testing Station* to determine the percentages of carbon dioxide and oxygen in atmospheres which suppressed combustion showed that the bonneted Wolf lamp burned until the atmosphere contained 3 per cent of carbon dioxide and 16.5 per cent of oxygen. Without the bonnet the same kind of lamp burned until there was 3 per cent carbon dioxide and 15.82 per cent oxygen. A candle flame was extinguished when the atmosphere in the bell jar contained 2.95 per cent of carbon dioxide and 16.24 per cent of oxygen. An acetylene lamp flame burned until there was 6.30 per cent carbon dioxide in the atmosphere and 11.7 per cent of oxygen, or until long after it became dangerous to human life. A natural gas flame issuing from a Bunsen burner went out when the bell jar contained 3.25 per cent carbon dioxide and 13.9 per cent oxygen.



PETTERSSON PALMQUIST APPARATUS
FIG. 7.

TABLE 10
THE HEAT EVOLVED BY ILLUMINANTS PER CANDLE-POWER PER HOUR (*Rubner*)

Illuminant	B.t.u. per Hr. per C.P.
Gas, ordinary split burner	300
Gas, Argand	200
Gas, Welsbach	31
Petroleum	100
Electric, incandescent	14
Electric, arc	4.3
Other authorities give the following:	
1 Gas jet 16 C.P.	3000 B.t.u. per Hr.
1 Welsbach	640 B.t.u. per Hr.
1 Arc, 1200 C.P.	3600 B.t.u. per Hr.
1 Incandescent 16 C.P.	160 B.t.u. per Hr.

The *vitiation of the atmosphere by manufacturing processes* is often very considerable as these processes may generate not only CO_2, but also poisonous gases and fumes, as well as give off

heat and water vapor, possibly in large quantities. In some cases dust is generated due to grinding, cutting or other operations, and in most states very stringent laws require an effective system of removal, so that the operatives will be protected. Such an exhaust system is described in detail at the end of this chapter.

Heating Due to Machinery, Lights, etc. Wherever *power* is used within a building *the heat equivalent of the work done* within the building will be set free and will amount to 2,545 B.t.u. per h.p. hr. or to 3,415 B.t.u. per kw. hr. This heat must be provided for in designing the ventilating system otherwise the building may be overheated.

Interior Temperatures. The *temperatures to be artificially maintained* within a heated building is considered in the chapter on "Heat Transmission of Buildings," and certain values are also indicated here in Table 11, for reference in ventilation work:

TABLE 11

USUAL INSIDE TEMPERATURES SPECIFIED

Public buildings	68° to 72° F.
Factories	65°
Machine shops	60° to 65°
Foundries, boiler shops, etc.	50° to 60°
Residences	70°
Bath-rooms	85°
Schools	70°
Hospitals	72° to 75°
Paint shops	80°

ANALYSIS AND EXAMINATION OF AIR

Characteristic Properties. The analysis or examination of air in order to ascertain its composition, and thereby the effectiveness of any system of ventilation, involves the determination of the following:

1. Carbon Dioxide and Monoxide.
2. Oxygen and Ozone, also Nitrogen.
3. Temperature and Humidity.
4. Dust and Bacteria.
5. Odors.

Carbon Dioxide. The determination of carbon dioxide, CO_2, may be made either by an *absolute or comparative method*. As already indicated this gas is not of itself injurious, but since it is probably the most typical product of respiration and combustion, it serves as *an index* of the extent to which the air has been vitiated by either of these two processes.

Moreover, although heavier than air, this gas will readily *diffuse* and mix uniformly with the air in the same way as the other more or less injurious gaseous products which usually accompany it. The amount of carbon dioxide generated by combustion and respiration is constantly on the increase, but the absolute amount contained in the air does not appear to increase, due to the fact that vegetation of all kinds possesses the property of assimilating the dioxide, retaining the carbon and giving off the oxygen as gas.

Pettersson-Palmquist Apparatus. The most *exact method* of ascertaining the *absolute amount* of CO_2 in the air was originated by *Otto Pettersson* and *A. Palmquist*, two European chemists. The results are said to be accurate to one part in 100,000, when an experienced operator is using the apparatus (Fig. 7), and the following description of its manipulation is given.

This apparatus has been slightly modified by *Dr. Rogers* to secure greater ease and positiveness in manipulation, but its principle of operation remains exactly as originally conceived and here illustrated.

The apparatus consists of a measuring vessel, *A*, connected with a U-shaped burette, *B* and *B'* from which communication can be made by a small stop cock, *b;* a manometer, *K*, marked

with a graduated scale nearly horizontal; and two stop cocks *f* and *g*, by means of which communication can be made with the outside air. One side of this manometer, *f*, is in communication with the closed vessel, *C;* the other side can be put in communication with the measuring vessel, *A*. The burette, *B*, contains a saturated solution of caustic potash (KOH). The flask, *E*, contains mercury, and by raising it when the stop cock *c* is open, the mercury will rise in the vessel, *A*, and the air will be driven out. If the flask, *E*, be lowered the mercury will flow from the measuring tube, and the amount of air entering *A*, can be measured by the graduations on scale, *S*. When the measuring tube, *A*, is full of air, the stop cocks *c*, *f*, and *g* being open, the position of the drop of liquid in the horizontal tube of the manometer, *K*, is accurately read. The stop cocks *c*, *a*, *f*, and *g* are then closed, that at *b* opened, and the vessel *E* raised, driving the air out of the measuring tube, *A*, and into the absorption burette, *B*. The caustic potash solution is brought to the mark *x* both before and after sending the air into the burette, *B*. This operation of raising and lowering the flask, *E*, is repeated several times; it is then lowered, and the air is drawn over into the measuring vessel; the cock, *a*, is then opened and the flask, *E*, manipulated until the reading of the manometer, *K*, on the horizontal scale agrees with that at the beginning of the test. The reading of the graduated tube, *A*, gives directly the amount of CO_2. The determinations are made with air of ordinary humidity, and there is a very slight correction due to this fact, which is not likely to equal, in any case, one part of CO_2 in one million parts of air. It will be noted that the burettes and measuring vessel are all surrounded by a water bath to prevent fluctuations in temperature during the determination.

Wolpert Air Tester. One of the most satisfactory *comparative methods* for ascertaining the *relative amount* of CO_2 in the air is that of *Wolpert's*. The apparatus consists of a glass cylinder about one inch in diameter by seven inches in length, closed at one end and provided with a rubber piston. The cylinder is graduated in cubic centimeters (c.c.) from the bottom upward to 50, and the hollow piston rod can be closed by a rubber cap. A solution of 1/20 gram of sodium carbonate (Na_2CO_3) in one liter of water, to which is added 0.075 gram of phenolphthalein, is used. This solution is pink so long as alkaline, but the color gradually disappears when CO_2 is added, until it finally disappears when the solution has been made neutral.

In operating the *Wolpert* air tester the piston is pressed down several times in order to drive the contents of the cylinder out through the hollow piston rod. After adding 2 c.c. of the above solution the piston is drawn out to the position marked 20 c.c., the hole in the rod is closed and the cylinder thoroughly shaken so that the CO_2 may be absorbed. If the solution is still pinkish the piston is pulled out a little further and the cylinder shaken as before. This operation must be continued until the solution is colorless. Tests are made both in the outside air as well as in the room in question.

Example. Suppose the piston stands at 46 c.c. with outside air and at 32 with room air. After deducting 2 c.c. from these readings for the volume of the original solution, we have that 44 c.c. of outside air contains the same amount of CO_2 as 30 c.c of room air, or the room air contains $44/30 = 1.46$ times as much CO_2 per c.c. as the outside air. On the usual assumption that the outside air contains 4 parts of CO_2 in 10,000 the air in the room would contain 5.84 parts in 10,000. The sodium carbonate solution must be carefully protected from the atmosphere after being prepared to prevent it from decomposing.

The *exact determination of oxygen, ozone and nitrogen* requires rather complicated apparatus for the manipulation of which one may refer to *A. H. Gill's* "Gas Analysis." The nitrogen of the air is never determined directly but is assumed to be the residue after removing all other gases from the sample under investigation.

Oxygen may be approximately determined by using an *Orsat* Apparatus (see chapter on "Fuels and Combustion") in which a sample of air can be drawn in, measured, and then passed into a burette containing a mixture of pyrogallic acid and caustic potash. The oxygen is absorbed, reducing the volume of gas by the amount of the quantity of oxygen present in the original sample. If either ozone or carbon dioxide were present in the above case, they would react like and be recorded as oxygen, unless removed in advance.

Temperature and Humidity. The determination of the temperature and humidity of the air is usually made simultaneously, by the use of *wet and dry bulb thermometers.* Very frequently these two thermometers are mounted together on a metal strip (see chapter on "Air Conditioning") the head of which is fitted with a handle by which the strip carrying the thermometers may be swung and made to revolve in a vertical plane. When arranged in this manner the apparatus is known as a *Sling Psychrometer,* and should be revolved at a peripheral velocity of not less than 15 ft. per second if consistent results are expected.

A more accurate form of this apparatus made in Germany is known as the *Assmann Aspiration Psychrometer* in which each thermometer is enclosed in a metal tube, and a small fan is used to draw air through the tubes at constant velocity. The bulbs of the thermometers are entirely protected from radiation. This causes a rapid circulation of air over the wet and dry bulbs, and the evaporation from the former abstracts sufficient heat from the bulb and air, provided the latter is not "saturated" with water vapor, to cause the wet-bulb thermometer to read lower than the dry. This difference, for any given dry-bulb temperature and a constant barometric pressure, increases as the actual amount of moisture per cu. ft. of the air decreases, and becomes a maximum for that dry bulb temperature and pressure when the air is absolutely dry (no water vapor present). The two thermometers read alike when the air carries all the vapor possible for that temperature and pressure, and the air is *saturated.*

At a given temperature a cu. ft. of space can hold a certain definite maximum amount of water vapor in suspension regardless of the presence or absence of other gases. If the space contains only one-half of the total possible amount for that dry-bulb temperature its *relative humidity* is 50 per cent, but its *actual* or *absolute humidity* is the weight in pounds of water vapor per cu. ft. of the space, or per pound of dry air. Air with a relative humidity less than 100 per cent can always take up more water vapor, but if air with a relative humidity of 100 per cent is cooled, even a small amount, some of the water vapor must condense. The temperature at which this condensation begins is called the *dew point.*

From tests made on air of known temperature and water vapor content, with corresponding definite vapor and barometric pressures, it has been possible to construct tables and curves from which relative and actual humidities can be read when the wet- and dry-bulb temperatures are known. (See "Psychrometric Chart" in chapter on "Air Conditioning.")

Example. If the dry-bulb reading is 75° and the wet-bulb 60°, it will be seen by reference to chart that the relative humidity is 41 per cent and the actual humidity is 0.0074 lb. of moisture per one pound of dry air and the "dew point" is 49°. One pound contains 7,000 grains.

Dust, Bacteria* and Odors. The *determination of dust* includes not only the amount actually present in a given volume of air, but an examination as to its character, such as abrasive, metallic, or fibrous. Soluble filters of pure sugar are used to screen out the dust particles and the sugar is then dissolved.

The *determination of bacteria* is carried out in a variety of ways; one very common method is to use tubes coated with beef jelly. The air in question as well as fresh outdoor air is drawn through separate tubes, and the number of colonies developing in the former case compared with the number developed in the latter case from an equal volume of fresh air. This comparison with the second or *control tube* gives a relative idea of the vitiation of the room air.

Where recirculation of air is practiced, as at the *Springfield* gymnasium, this bacterial examination is of considerable importance.

The *determination of odors* must depend largely on the keenness of smell of the investigator. Odors may in themselves be quite harmless, but as an index of contamination may prove of great importance.

It must always be remembered that air which is "fit" according to all other ventilation standards may prove absolutely intolerable due to the presence of some disagreeable odor.

*NOTE.—For other "Methods of Bacterial Analysis of Air" see *Bulletin No. 409,* August, 1915, by *G. L. A. Ruehle and W. L. Kulp, New York Agricultural Experiment Station,* Geneva, N. Y.

Ozone. Use has been made of *ozone* to mask or destroy odors, and with proper concentration, in the presence of moisture some success has been achieved. Where people are present, it is not advisable to use ozone in concentrations much above 3 parts in one million, although from 1 to 6 parts are used.

VENTILATION LAWS

The *ventilation laws* have increased very rapidly in the last few years, not only as regards the number of states which have added such laws to their code, but also as to the scope and effectiveness of these statutes. In many cases a special ventilation officer or commission has been appointed to see to the enforcement and extension of the requirements for compulsory ventilation, so that it behooves the architect or engineer to become thoroughly familiar with the law of the state or states wherein he practices.

The more important provisions of the laws of several states are given herewith and a summary of the law recently enacted by the legislature of the state of *Ohio* is given at some length as an example of the regulations with which the architects and engineers must conform in preparing plans and specifications.

TABLE 12

VENTILATION LAWS OF VARIOUS STATES FOR SCHOOL BUILDINGS

(Minimum Requirements)

State	Cu. Ft. Fresh Air per Hour per Pupil at 70°	Cu. Ft. or Space per Pupil	Sq. Ft. of Floor Space per Pupil	Glass Area in per Cent of Floor Space	Temp. of Room in Coldest Weather F°	Min. Height of Ceiling, Ft.
Massachusetts*	1800	...	..	..	70	.....
New Jersey	1800	200	12	20	..	12′–0″
New York	1800	200	15	..	..	.....
Pennsylvania	1800	200	15	25	70	.....
Vermont†	1800	200 to 300	20	..	70	12′–0″
Indiana	1800	225	..	..	70	.....
Ohio‡	6 Changes per Hr.	... ...	.. ..	.. ..	70 ..	....
Maine	2000	240	..	..	..	.....
Minnesota	1800	216	18	..	..	12′–0″
Louisiana	1800	200	..	14	..	.....

NOTE.—1800 cu. ft. = 135 lb. of air at 70° F.
In all cases a positive means of removing the foul air is required.
* The *Massachusetts* law further requires:
1. That the apparatus will, with proper management, heat all the rooms including corridors, to 70 degrees in any weather.
2. That with the rooms at 70° and a difference of not less than 40° between the temperature of the outside air and that of the air entering the room at the warm-air inlet, the apparatus will supply at least 30 cubic feet of air per minute for each scholar accommodated in the rooms.
3. That such supply of air will so circulate in the rooms that no uncomfortable draft will be felt, and that the difference in temperature between any two points on the breathing plane (5 feet), in the occupied portion of a room will not exceed 3°.
4. That vitiated air in amount equal to supply from inlets will be removed through the vent ducts. This law is enforced by the *Massachusetts District Police.*
† The *Vermont* requirements apply to all public buildings and include churches, hotel buildings more than two stories high and places of amusement more than one story high; also factories, mills and workshops more than two stories high in which persons are employed above the second story.
‡ The *Ohio* law requires that corridors, hallways, playrooms, toilets, assembly rooms, gymnasiums, and manual training rooms shall be heated to 65° and all other parts of the building to 70° in zero weather.

The *Illinois* ventilation law for factories, mercantile establishments, mills and workshops requires that a reasonable and equitable temperature consistent with the reasonable requirements of the manufacturing process be maintained, and that no unnecessary humidity which would jeopardize the health of employees will be permitted. At least 500 cubic feet of space

shall be provided per person in buildings and rooms using lights that consume oxygen, and at least 250 cubic feet for buildings and rooms using electric lights.

All rooms having at least 2,000 cubic feet of space per person and having at least an outside door and window area of one-eighth of the floor space shall not be required to have artificial means of ventilation. All rooms having less than 2,000 cubic feet of space per person and more than 500, and with an outside door and window area as above specified, must be provided with an artificial means of ventilation at the rate of 1,500 cu. ft. of fresh air per person per hour during such weather when windows cannot be left open for ventilation.

When the outside door and window area is less than as specified, then artificial ventilation must be supplied at the rate of 1,800 cu. ft. per person per hour.

The *New York* factory ventilation law requires that in each workroom proper and sufficient means of ventilation shall be employed. If excessive heat be created, or if steam, gases, vapors, dust and other impurities that may be injurious to health, be generated in the course of the manufacturing process carried on therein, the rooms must be ventilated in such a manner as to render them harmless as far as possible. It will be readily apparent that these requirements are entirely too general to secure satisfactory results unless vigorously enforced by competent inspectors.

The *Ohio* law, which follows, as well as the law of *Massachusetts*, attempts to provide very definite regulations for heating and ventilating all classes of buildings. Future legislation in other states will undoubtedly take a more specific form, establishing complete and definite codes for the heating and ventilation not only of public buildings but of workshops, factories, and mercantile establishments as well.

REQUIREMENTS OF THE DEPARTMENT OF INSPECTION OF THE INDUSTRIAL COMMISSION OF OHIO FOR THE HEATING AND VENTILATION OF PUBLIC BUILDINGS, HOSPITALS, ASYLUMS AND HOMES.

Temperature

A heating system shall be installed which will uniformly heat the various parts of the building to the following temperatures in zero weather:

Theatres and Assembly Halls. All parts of the building, except storage rooms, to 65° F.

Churches. Auditorium, social and assembly rooms, 65° F.

All other parts of the building, except storage rooms, to 70° F.

School Buildings. Corridors, hallways, play rooms, toilets, assembly rooms, gymnasiums and manual-training rooms, 65° F.

All other parts of the building to 70° F.

Hospitals, Asylums and Homes. Operating rooms, 85° F.

All other parts of the building, except storage rooms, to 70° F.

Change of Air

The heating system shall be combined with a system of ventilation which at normal temperature will change the air the following number of times or supply to each person the following number of cubic feet of air per hour.

Theatres. Parlors, retiring, toilet and check rooms, six changes per hour.

Auditoriums, 1,200 cu. ft. of air per person per hour.

Assembly Halls. When used in connection with a school building, lodge building, club house, hospital or hotel, six changes per hour; and in all other assembly halls, 1,200 cu. ft. of air per hour per person.

Churches. Auditoriums, assembly rooms and social rooms six changes per hour.

School Buildings. All parts of the building, except corridors, halls and storage rooms, six times per hour.

Asylums, Hospitals and Homes. (Rooms with fixed capacity.)

	Adult	Children	Babies
Hospitals, contagious and epidemic	6,000	4,000	3,000
Hospitals, surgical and medical	3,000	2,400	1,500
Penal Institutions	1,800	1,800	.....
All other buildings	1,800	1,500	.....

Rooms with variable capacities:

Hospitals, contagious and epidemic	12 times per hour
Hospitals, surgical and medical	12 times per hour
All other buildings	6 times per hour

Rooms accommodating four or less persons need not be provided with a system of ventilation.

Radiators

No radiator shall be placed in any aisle, foyer or passageway of a new theatre, assembly hall or church, but such radiators may be placed in recesses in the walls.

Registers

No floor registers shall be used in theatres, assembly halls, or hospitals.

No floor registers, except foot warmers, shall be used in a school building.

Floor registers may be used in churches.

Otherwise all vent registers shall be placed not more than 2 in. above the floor line, and warm air registers not less than 8 ft. above the floor line (except when such registers are used when a change of air is not prescribed).

Systems to be Installed Where a Change of Air is Required

The system to be installed when a change of air is required shall be either a gravity or mechanical furnace system, gravity indirect steam or hot-water system; mechanical indirect steam or hot-water system, or split steam or hot-water system, except in hospitals where a direct-indirect system may be used in connection with an exhaust fan.

The fresh air supply shall be taken from outside the building and no vitiated air shall be reheated.

All vitiated air shall be conducted through flues or ducts and be discharged above the roof of the building.

Exceptions. Standard ventilating stoves may be used in the following buildings:

Assembly halls seating less than 100 persons.

Churches seating less than 100 persons.

All school buildings, hospitals, asylums and homes.

Furnaces

Furnaces may be used in all classes of buildings.

Gravity Indirect Hot-Water or Steam Radiator Systems

Indirect hot-water or steam radiators shall be located in basement fresh-air rooms directly at the base of masonry hot air flues, and shall be properly connected to same with galvanized iron housing.

Indirect Radiating Surface for Heating and for Ventilating Purposes

One square foot of radiating surface shall be provided to heat not more than the following number of cubic feet of air per hour:

Height	Steam	Hot Water
First story	200	125
Second story	250	160
Third story	300	200
Fourth story	250	235

For Heating Wall and Glass Surfaces. The amount of radiating surface for the heating of the glass and wall surface shall not be less than that obtained by adding together the glass surface and one-fourth the exposed wall surface, both in square feet, and multiplying by the following factors:

Height	Steam	Hot Water
First story	0.7	1.05
Second story	0.6	0.9
Third story	0.5	0.75
Fourth story	0.4	0.5

Accelerating or Aspirating Coils for Vent Flues. Vent flues used in connection with a gravity indirect steam or hot-water system shall be provided with accelerating coils placed 1 ft. above the vent openings.

Mechanical Fan Plenum System

This system shall be designed with furnaces, tempering coils or blast coils so as to furnish heated air, to have cleaning screens, fan plenum chamber, galvanized iron or masonry horizontal ducts, masonry hot air flues, electric motor, gas or gasoline engine, or a low pressure steam engine operating on a steam pressure not to exceed 35 lb. gage to operate fan and such other device as is necessary to make this a complete working system. All parts and apparatus in connection with the installation is to be of ample size to make a perfectly free and easily working system, and must thoroughly heat all portions of the building without forcing.

Velocity of Air

The velocity of the air traveling through ducts, flues, etc., shall never exceed the following number of feet per minute:

Ducts, Flues, etc.	Feet per Minute
Fresh air screens (small mesh)	600
Fresh air ducts, gravity system	300
Fresh air ducts, mechanical system	850
Tempering coils, gravity system	300
Tempering coils, mechanical system	1,000
Furnaces, gravity system	400
Furnaces, mechanical system	900
Trunk ducts, mechanical system	1,000
Laterals, branches and single ducts, mechanical system	750
Vertical flues, mechanical system	500
Vertical warm air flues, gravity system, first story	300
Vertical warm air flues, gravity system, second story	350
Vertical warm air flues, gravity system, third story	390
Vertical vent flues less than 20 ft. high	300
Vertical vent flues 20 to 33 ft. high	350
Vertical vent flues 33 to 46 ft. high	390
Vertical vent flues 46 to 60 ft. high	440
Warm air registers	300
Vent registers	300

Maximum Speed of Fans

The maximum speed of fans used in connection with either an exhaust or plenum system of heating or ventilating, under normal conditions shall never exceed the following:

Diameter of Fan in Inches:	18	24	36	48	60	72	96	120	180
Revolutions per minute:	700	550	400	300	225	175	150	125	75

Location of Heater Room

No heater room shall be located under the auditorium, stage, lobby, passageway, stairway or exit of a theatre; nor, under any exit, passageway, public hall or lobby of an assembly hall, church, school building, asylum, hospital or home. This applies to new buildings, and a changed location of a heater room in an existing building.

No cast-iron boiler carrying more than 10 lb. pressure or steel boiler carrying more than 30 lb. pressure shall be located within the main walls of any school building.

Standard Fireproof Heater Room for New Buildings

All furnaces and boilers, including the breeching, fuel rooms and firing spaces, shall be enclosed by brick walls not less than 12 in. thick, or by monolithic concrete walls not less than 8 in. thick; and the ceiling over the same shall not be less than the following: reinforced concrete slab 4 in. thick, brick arches 4 in. thick, covered with 1 in. of cement mortar and supported by fireproof steel with the necessary tie rods, or by hollow tile arches 6 in. thick covered with 2 in. of concrete, plastered on the under side and supported by fireproof steel with the necessary tie rods.

All openings in the above apartments from the other parts of the building shall be covered by standard self-closing fire doors.

Heater Rooms for Old Buildings

In old buildings, the boiler or furnace and fuel rooms shall be enclosed in the same masonry walls and shall have standard fire doors on opening to same, and the entire ceiling shall be fireproofed as follows: First overlay the entire ceiling with 1/4 in. asbestos board, lapped at least 1½ in. at joints, then furr same with 1½ in. high metal furring spaced 12 in. on centers; then lath with metal lath and heavily plaster with asbestos and Portland cement plaster.

SPECIAL EXHAUST SYSTEMS IN COMPLIANCE WITH STANDARD VENTILATION LAWS FOR DUST REMOVAL

The removal of dust created in grinding and buffing operations must be suitably provided for in accordance with definite statutory requirements in many states. The following data and specifications which conform to the New York state law have been prepared by *Wm. Newell*, of the *Department of Labor* of that state, and give the proper proportions and details for systems of varying capacities as indicated in the tables, which were taken from the *Practical Engineer* for May 15, 1913:

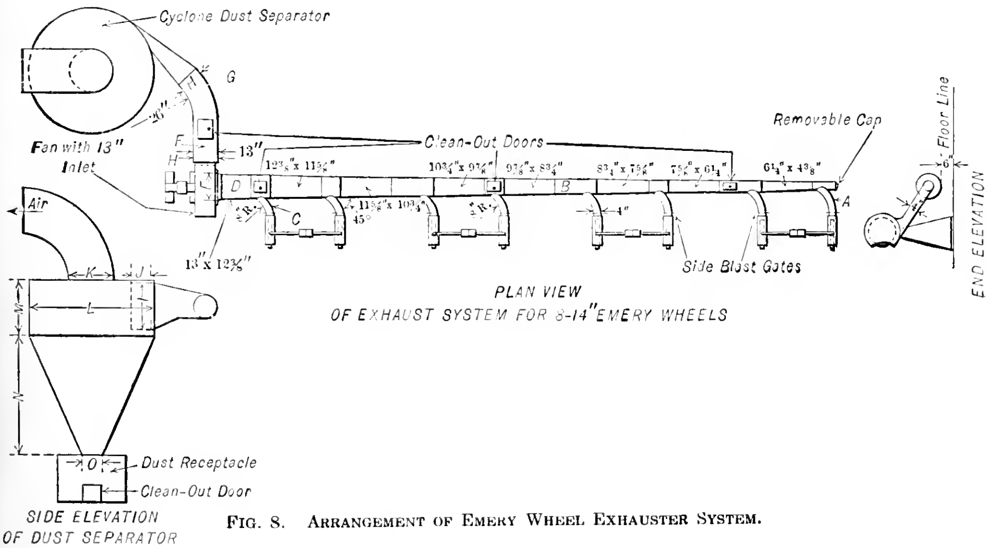


FIG. 8. ARRANGEMENT OF EMERY WHEEL EXHAUSTER SYSTEM.

"**General Arrangement.** In the general layout, it is recommended that emery-wheel and buffing-wheel systems be kept separate, because sparks from the emery wheels may set fire to the lint and dust from the buffing wheels if both are carried through the same suction main.

"In those wheels which have the top running toward the operator, the main suction duct should be back of and below the wheels, not less than 6 in. above the floor, to avoid possible charring in case of fire in main duct, and also to permit sweeping under it. This location is recommended so that all dust will settle into the main.

"Main suction and discharge pipes should be made short and with as few bends as possible, to cut down the friction loss, and where there must be considerable distance from the wheels to

See Appendix for a description of the Gale Centrifugal Dust-Collecting System in which a collector of the air-washer type is used.

the cyclone, it is best to keep the fan near the last branch which enters the main, making the discharge end the long one.

"Pockets or low places where dust may accumulate should be avoided, if possible, and if this cannot be done, a trap should be installed and kept cleaned out.

"**Branches and Mains.** Branch pipes must enter the main suction duct at an angle not exceeding 45 deg., and must incline in the direction of the air flow at the junction with the main as shown in the illustration herewith. The ends of branch pipes must not project into the main duct, and the duct must be made so that all laps in piping point in the direction of air flow.

"For bends, turns or elbows in main or branch pipes, the radius in the throat must be at least 1.5 times the diameter of the pipe, and should preferably have the radius twice the diameter of the pipe wherever space permits. These branches must enter the main on the top or sides, never at the bottom, and two branches should never enter a main exactly opposite each other.

"The main suction should be enlarged between every two branch pipes entering it, if possible, and in no case should more than two branches enter the main duct in a section of uniform area. All enlargements in the main must be made on a taper, not by abrupt change (Fig. 8).

"Each branch pipe should have a shut-off damper which may be closed when the wheel is not in use; but not more than one-fourth of these gates should be closed at any one time, as the air velocity in the main pipe might drop so low as to let dust accumulate on the bottom.

"Elbows should be made of metal, one or two gages heavier than the main pipe, because the wear is greater on them.

"**Hoods.** In forming the hood for the wheel, it should go far enough forward beneath the front to insure that all dust will enter it, even though considerable space must be left between the wheel and the lower part of the hood in order not to interfere with the work. The branch pipe should lead from the hood as nearly as possible at the point where dust will naturally be thrown by the wheels, and no screen should ever be used across the mouth of the branch pipe, as this will obstruct the passage of material and may, in a short time, be almost stopped up.

"A trap at the junction of hood and branch pipe is desirable to collect the heavier particles of dust, thus taking some wear off the fan, and also to catch nuts or small articles dropped by accident into the hood.

"**Sizes of Pipes.** The minimum sizes of branch pipes allowed for different sizes emery or other grinding wheels are shown in Table 13, and also sizes of branch pipes for buffing or polishing wheels. In case of a wheel thicker than given in the table, it must have a branch pipe at least as large as called for by its grinding surface.

TABLE 13

MINIMUM SIZES OF BRANCH PIPES FOR GRINDING, POLISHING AND BUFFING WHEELS

	Emery and Other Grinding Wheels		Buffing, Polishing or Rag Wheels	
	Maximum Grinding Surface, Sq. In.	Minimum Diameter of Branch Pipe, Sq. In.	Maximum Grinding Surface, Sq. In.	Minimum Diameter of Branch Pipe, Sq. In.
6 inches or less, not over 1 inch thick	19	3	19	3½
7 to 9 inches, not over 1½ inches thick	43	3½	...	...
7 to 12 inches, not over 1½ inches thick	...	...	57	4
10 to 16 inches, not over 2 inches thick	101	4	...	...
13 to 16 inches, not over 2 inches thick	...	...	101	4½
17 to 19 inches, not over 3 inches thick	180	4½	...	...
17 to 20 inches, not over 3 inches thick	...	...	189	5
20 to 24 inches, not over 4 inches thick	302	5	...	...
21 to 24 inches, not over 4 inches thick	...	...	302	5½
25 to 30 inches, not over 5 inches thick	472	6	472	6½

"These branch pipes must have sizes not less than specified in the table, throughout their entire length, and the area of the main must be so chosen that it shall be at least 20 per cent greater than the combined areas of branch pipes entering it between any given point and the dead end of the system.

"The inlet of fan or exhauster shall be at least 20 per cent greater in area than the sum of the areas of all branch pipes. Table 14 gives the size of the main suction duct at any point, for any number of uniform size branch pipes with 20 per cent excess of area.

TABLE 14

SIZE OF MAIN SUCTION DUCT FOR ANY NUMBER OF UNIFORM BRANCHES, BASED ON TWENTY PER CENT EXCESS AREA

Number of Branch Pipes	Diameter of Branch Pipe, in Inches								
	3	3½	4	4½	5	5½	6	6½	7
	Area of Branch Pipe, in Square Inches								
	7.07	9.62	12.57	15.90	19.64	23.76	28.27	33.18	38.49
	Area of Branch Pipe Plus 20 Per Cent, in Square Inches								
	8.48	11.54	15.08	19.08	23.56	28.51	33.93	39.82	46.18
1	3⅜	3⅞	4⅜	5	5½	6	6⅝	7⅛	7¾
2	4¾	5½	6¼	7	7¾	8⅝	9¼	10⅛	10⅞
3	5¾	6⅝	7⅝	8⅝	9½	10½	11½	12⅜	13¼
4	6⅝	7¾	8¾	9⅞	11	12⅛	13⅛	14¼	15⅜
5	7⅜	8⅝	9⅞	11	12¼	13½	14¾	16	17⅛
6	8⅛	9½	10¾	12⅛	13½	14¾	16⅛	17½	18¾
7	8¾	10¼	11⅝	13⅛	14½	16	17½	18⅞	20¼
8	9⅜	10⅞	12⅜	14	15½	17⅛	18⅝	20⅛	21¾
9	9⅞	11½	13⅛	14⅞	16½	18⅛	19¾	21⅜	23
10	10½	12⅛	13⅞	15⅝	17⅜	19⅛	20¾	22½	24¼
15	12¾	14⅞	17	19⅛	21¼	23⅜	25½	27⅝	29¾
20	14¾	17⅛	19⅝	22⅛	24½	27	29½	31⅞	34⅜
25	16½	19¼	22	24¾	27½	30⅛	32⅞	35⅝	38⅜
30	18	21	24	27	30	33	36	39	42

"The discharge pipe from the fan shall be as large or larger than the fan inlet throughout its entire length. Main trunk lines for both suction and discharge shall have suitable clean-out doors not over 10 ft. apart, and the dead end of the suction main shall have a removable cap on it. A draft head of 2 in. of water shall be maintained in each branch pipe within one foot of the hood, this draft being tested by a U-tube water gage with rubber connection.

"**The Cyclone.** *Mr. Newell* states that recommendations for the cyclone separator, or dust collector, are difficult, as the separator must be proportioned to suit operating conditions. Light dusts require a larger separator than the heavy dusts. Table 15 gives dimensions of one prominent manufacturer, such as are suitable for metallic dusts and wood shavings. For these the inlet area of the cyclone should be as large as the area of the discharge pipe from the fan. For light buffing dusts the air outlet from the cyclone should be large enough so that the discharge velocity will not exceed 480 ft. a minute, and 300 ft. is better. Other dimensions of the separator should then be made proportionately. The air outlet should have a proper elbow to exclude the weather, but be otherwise unobstructed. The cyclone should be set so that there is ample clearance beneath it, and dust should never be allowed to pile up as high as the bottom of the separator.

"The illustration herewith shows a sample system for 8 14-in. emery wheels with dimensions."

TABLE 15

DIMENSIONS OF CYCLONE TYPE OF DUST SEPARATORS

Diam. of Fan Outlet, in Inches	Area of Fan Outlet, in Square Inches	Openings in Separator					Dimensions of Separator			Shipp'g Weight, in Pounds
		Inlet		Air Outlet		Dust Outlet	Outside Diam. of Cylinder, in Inches	Height of Cylinder, in Inches	Length of Cone, in Inches	
		Size, in Inches	Area, in Square Inches	Diam., in Inches	Area, in Square Inches	Diam., in Inches				
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)
H		*JxI*		*K*		*O*	*L*	*M*	*N*	
5........	20	2½x 9	23	8½	56	3	29½	14	26½	70
7 or 8...	38 or 50	3½x13	47	13	132	6	41½	18½	37½	140
10.......	78	5 x18	90	17	227	6	53½	23	50	245
11 or 12.	95 or 113	5½x21	115	20	314	10	59½	26	56	315
13 or 14.	133 or 154	6½x24	156	23½	433	10	65½	29	61½	395
16 or 17.	201 or 227	8 x30	240	28	615	10	77½	35	72½	575
18.......	254	8½x32	272	31	754	10	83½	38	77½	715
19 or 20.	283 or 314	9 x35	315	33	855	10	89½	41	82½	875
22.......	380	10 x41	410	39	1,194	10	97½	47	89	1,000
23 or 24.	415 or 452	10½x43	451	41	1,320	11	101½	49	93	1,095
26.......	531	11 x48	528	46	1,662	12	109½	54	99½	1,600
28.......	621	11½x54	621	52	2,123	12	117½	60	109½	1,855
30.......	707	12 x60	720	58	2,642	12	125½	66	115½	2,155
31 or 32.	754 or 804	12½x63	807	61	2,922	13	129½	69	118½	2,250
34.......	908	13½x69	932	67	3,525	13	137½	75	126½	2,555
36 or 37.	1,017 or 1,075	14½x75	1,087	73	4,185	14	145½	81	133½	2,900
38.......	1,134	15 x78	1,170	76	4,536	14	149½	84	137½	3,065
39 or 40.	1,194 or 1,256	15½x81	1,255	79	4,901	14	153½	87	141½	3,235
41.......	1,320	16 x84	1,344	82	5,281	14	157½	90	145½	3,395

NOTE.—Certain intermediate sizes for Fan Outlet have been omitted from Column No. 1. Separators are made in all whole sizes from 5 to 41 inches.

CHAPTER XIII

GRAVITY-INDIRECT HEATING BY STEAM AND HOT WATER

GENERAL DESCRIPTION OF SYSTEM

Method of Operation. Heating by means of *gravity-indirect radiators*, which are usually supplied with cold air from out of doors, furnishes a very satisfactory means of both providing for the heat loss from a room and supplying air for ventilation at the same time. The radiators in this system (Fig. 1) are usually placed in the basement, properly encased, and are of a special type designed with extended surfaces and intended to give off their heat by convection to the air passing over them. The heated air then rises through vertical flues, due to its diminished density,

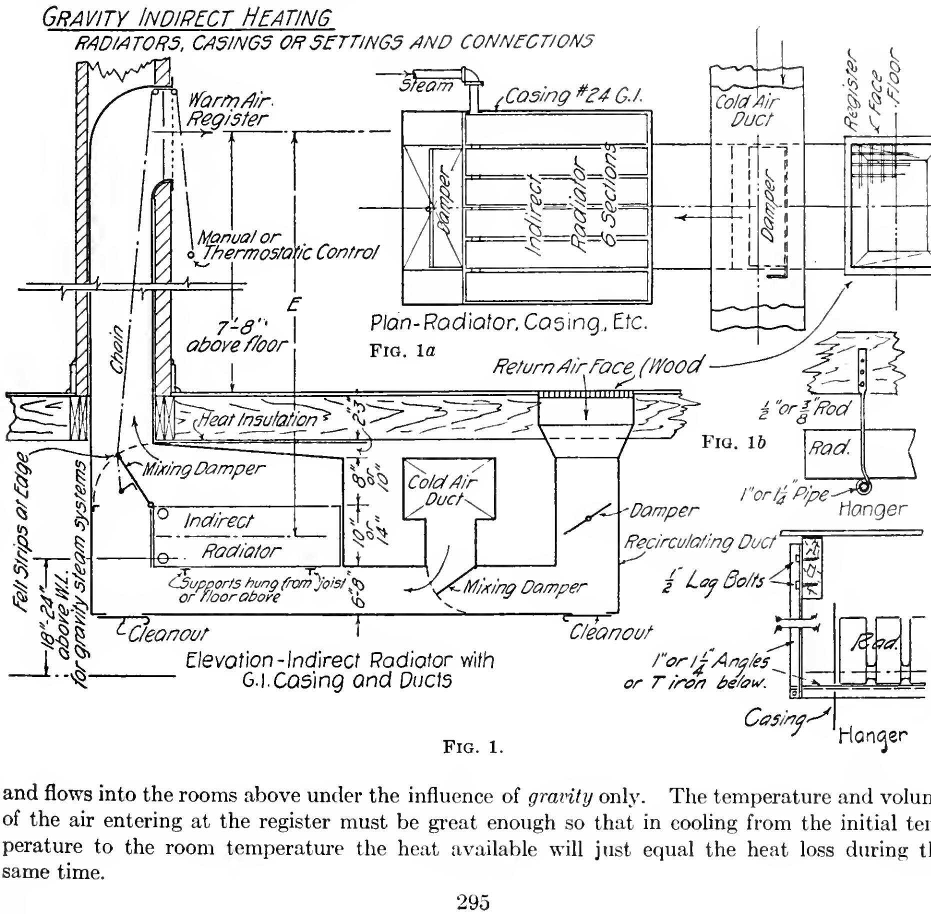


Fig. 1.

and flows into the rooms above under the influence of *gravity* only. The temperature and volume of the air entering at the register must be great enough so that in cooling from the initial temperature to the room temperature the heat available will just equal the heat loss during the same time.

Ventilation an Inherent Feature. In cases where *ventilation is a requirement* the air volume required for this purpose may become so large that the entering air temperature will be but slightly above the room temperature.

It is not only necessary to provide ducts for supplying air to each room heated, but in order to establish and maintain a constant flow provision must be made for *positively removing the air* in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts which take air at or near the floor line.

This system is very similar in principle to the ordinary gravity warm-air furnace system, except that it uses steam or hot water as the heating medium, confined in suitable radiators, instead of hot furnace gases. It differs from the hot-blast system in that no fan or blower is used to force the air through the system, as the air flow is maintained *entirely by gravity* or natural draft conditions. Since this *gravity head* is very slight, it is necessary to make all ducts as short as possible, especially the runs from the indirect radiators to the base of the vertical hot-air flues.

Data Necessary for Proportioning the System. In order to design a gravity indirect heating system it is necessary to know: (1) the heat loss from each room, (2) the quantity of air to be supplied, either for heating only, or for heating and ventilation, and the temperature at which it must enter the room, (3) the probable flue velocities for air rising through vertical flues running to the various floors above the indirect radiator, from which the areas of ducts and registers may be found, (4) the heat transmitting efficiency of the indirect radiator for air flowing at such a rate as will give the required temperature rise, from which the size of the indirect radiator and its casing may be determined, and (5) the allowable natural draft velocities in vent flues, for determining the size of the air removal ducts.

THE DESIGN OF A GRAVITY-INDIRECT SYSTEM

Heat Loss. The heat loss from each room in B.t.u. per hour must be determined for the particular inside and outside temperature conditions existing during the most severe weather which will occur in the locality where the plant is placed. For methods of calculating this loss see the chapter on "Heat Transmission of Building Materials."

Quantity of Air Required. The quantity of air required per hour depends on the temperature at which the air is to enter the room, the room temperature, usually 70° F., and the heat loss to be supplied. The entering air temperature is ordinarily assumed, and is taken, from 5° to 10° less than the temperature of the air leaving the indirect radiator, due to heat losses from the ducts connecting the radiator and the register. The temperature of the air leaving the radiator depends on (1) the temperature of the steam or water, (2) the entering air temperature, and (3) the velocity of flow through the free area of the radiator, as explained in this chapter, under "Performance of Gravity Indirect Radiators." The temperatures generally assumed, when air enters the radiator at zero with steam at 2 lb. gage, are 120° F. for steam and 100° F. for hot water, with average water temperature at 170° F. Indirect radiators are ordinarily made one section in depth to give these temperatures.

The quantity of air required in pounds per hour is:

$$W = \frac{H}{0.24\,(t_2 - t_1)}$$

and expressed in cu. ft. at temperature at register, $Q_2 = W/d_2$, in which

W = pounds of air to be supplied per hour.

Q_2 = cu. ft. of air to be supplied per hour at temperature t_2.

H = heat loss from room in B. t. u. per hour.

t_2 = temperature of air at the register (assumed or determined by ventilation requirements).

d_2 = density of the air at temperature t_2.

t_1 = room temperature desired (70°F. usually).

0.24 = specific heat of air.

Area of Warm-Air Duct. The area of warm-air duct required to supply the air depends on the *velocity* in same, which in turn depends on the height of warm-air register above the center of indirect radiator, and on the average temperature in the warm air flue. The theoretical velocities (Table 1) may be calculated, as explained under the chapter on "Drafts and Chimneys," where it was shown that:

$$V = 8.02\sqrt{\frac{h\,(t_c - t_o)}{460 + t_o}}, \text{ in which}$$

V = theoretical velocity in feet per sec. (no friction).
h = height in ft. from center of radiator to center of register, or elevation (E) in Fig. 1.
t_c = average temperature of warm air column.
t_o = outside air temperature, when outside air is used.

TABLE 1

THEORETICAL VELOCITY OF (V) AIR, IN FEET PER SECOND, DUE TO NATURAL DRAFT

(*R. C. Carpenter*)

Height of Flue in Feet	Excess of Temperature in Flue Above External Air								
	5°	10°	15°	20°	25°	30°	50°	100°	150°
1	0.8	1.1	1.4	1.6	1.8	2.0	2.5	3.6	4.4
5	1.8	2.5	3.1	3.6	4.0	4.5	5.6	8.1	9.9
10	2.6	3.6	4.4	5.1	5.7	6.6	8.1	11.4	14.0
15	3.1	4.4	5.4	6.3	7.0	7.7	9.9	14.0	17.1
20	3.6	5.1	6.3	7.2	8.1	8.8	11.4	16.1	19.8
25	4.0	5.7	7.1	8.1	9.0	9.9	12.8	18.0	22.1
30	4.4	6.3	7.8	8.8	9.9	10.8	14.0	19.8	24.2
35	4.8	6.8	8.4	9.5	10.7	11.7	15.1	22.3	26.1
40	5.1	7.3	8.9	10.2	11.4	12.5	16.1	22.8	27.9
45	5.4	7.7	9.4	10.8	12.1	13.3	17.1	24.2	29.6
50	5.7	8.1	9.9	11.4	12.8	14.0	18.0	25.5	31.1
60	6.3	8.8	10.8	12.6	14.0	15.3	19.8	27.8	33.3
70	6.8	9.5	11.7	13.6	15.2	16.5	21.4	30.0	36.1
80	7.3	10.2	12.5	14.4	16.2	18.7	22.9	32.2	38.9
90	7.7	10.8	13.3	15.3	17.2	18.8	24.3	34.2	41.6
100	8.1	11.4	14.0	16.2	17.8	19.8	25.6	36.0	45.2
125	9.1	12.8	15.6	18.1	20.1	22.1	28.7	40.3	49.3
150	9.9	14.0	17.2	19.8	22.2	24.3	31.4	44.3	54.3

The *actual velocities* V_1, V_2 and V_3 for the various floors will seldom exceed one-third of the theoretical value as calculated above, and in general the following values (Table 2) are assumed for steam and water systems, with registers placed 7′-0″ above the floor:

TABLE 2

AIR VELOCITIES IN WARM AIR FLUES—GRAVITY-INDIRECT (OUTSIDE AIR 0° F.)

Number of Floor	Elevation (E) Center Register to Center Radiator	Air Velocity in Feet per Minute	
		Steam (120° F.)	Hot Water (100° F.)
	Feet		
In 1st Floor	3–5	180	160
1*	10	240	200
2*	20	360	300
3*	30	420	360

* Note.—Inlet register 7′-0″ above floor.

Having determined the allowable velocity the area of the duct is found as follows:

$$A = \frac{Q_2}{60 \times V_a}, \text{ in which}$$

A = area of warm air duct or flue in sq. ft.

Q_2 = cu. ft. of air to be supplied per hr. at temperature t_2 at register. The average temperature in the flue is slightly greater than t_2

V_a = actual air velocity in feet per min. for 1st, 2nd, or 3rd floor.

The *cold-air ducts* are made about 80% of the area of the warm-air ducts, and in case of *recirculation* of air from the room through the radiator the recirculating ducts are usually made the same size as the warm-air supply ducts.

Warm-air ducts are often proportioned by allowing a certain number of sq. in. area in same for each sq. ft. in the indirect radiator. This ratio should vary with the height of duct. Arbitrary values often assumed are 2 sq. in. per 1 sq. ft. for steam and 1½ sq. in. per 1 sq. ft. for hot water.

Radiating Surface Required. The amount of radiating surface required depends on the heat transmitting efficiency of the indirect radiator, or its coefficient of transmission K, as already stated. Thus the heat given off by 1 sq. ft. of indirect surface is:

$$R = K\left[t_s - \frac{(t_3 + t_o)}{2}\right]$$

and the heat to be supplied the air passing the radiator per hour is:

$$H_1 = 0.24 \times W \times (t_3 - t_o), \text{ from which}$$

the amount of radiation may be found as follows:

$$S = \frac{H_1}{R}, \text{ in which}$$

R = B.t.u. given off by 1 sq. ft. of radiation per hour.

K = B.t.u. given off by 1 sq. ft. of radiation per hour per 1° difference between the steam or water and average temperature of the air passing radiator.

t_s = temperature of steam, or average water temperature.

t_3 = temperature of air leaving the radiator.

t_o = temperature of air entering radiator.

H_1 = B.t.u. required at the radiator to heat all the air (W pounds per hour) from t_o to t_3 degrees.

S = sq. ft. (net) of heating surface having a coefficient K.

See "Performance of Indirect Radiators" for values of K.

Ventilation a Requirement. If ventilation requirements *exceed the amount of air required for heating only*, then more air at a lower temperature must be supplied. This condition is determined by comparing the air required for ventilation (based on some standard, such as 1,800 cu. ft. $\times N$, where N is the number of occupants) with the volume necessary for heating, measured at room temperature t_1 and density d_1. Thus, should $1{,}800 \times N$ be greater then W/d_1 then W must be increased until $W_v = 1{,}800\, d_1\, N$. Since we are now going to supply more air than our assumed register temperature demands, it will be necessary to supply this air at a lower temperature than that assumed to prevent overheating.

Then, since the fixed heat loss is,

$$H = 0.24\, W_v\, (t_2' - t_1)$$

that
$$t_2' = \frac{H + 0.24\, W_v\, t_1}{0.24\, W_v}, \text{ in which}$$

t_2' = new temperature of air leaving register (lower than t_2).
W_v = lb. of air required per hour (greater than W).

The new temperature at the radiator is $t_3' = t_2' + (5° \text{ to } 8°)$, and it must be used in finding the amount of radiating surface, instead of t_3, in order to satisfy the new conditions.

Ventilating Flues. The ventilating flues required for gravity-indirect heating may be of the simple *gravity type*, or of the *aspirating type*, in which latter case a heating coil is placed in the flue in order to accelerate the flow.

These flues should not be proportioned as some function or percentage of the heating flue, such as 75% of the area of the latter, since the vent flues from the first floor are much longer than the heat flues to this same floor. In the case of upper floors the reverse is true. The *size* of gravity vent flues depends entirely on the velocity of outflow due to height of flue and temperature differential. This velocity will steadily diminish as the outside temperature rises until with the inside and outside temperatures the same the velocity will become zero in simple gravity flues, while in aspirating flues the heating coil will always cause a temperature differential to exist and hence a positive outflow.

It is generally customary to assume a minimum temperature difference of at least 20° for which a gravity flue should be proportioned, using about 50% of the theoretical velocities for the given height, as stated in Table 3. These flues are generally straight and offer much less friction than the heat flue, which must overcome the additional friction of radiator and cold-air duct.

TABLE 3

VELOCITIES IN VENT FLUES

Height of Flue Above Coil or Center of Register in Feet	Velocities of Air in Feet per Min. for Temperature Difference of 20° F.	
	No Friction	Actual; Reduced 50% for Friction
10	306	153
20	432	216
30	528	264
40	612	306
50	684	342
60	756	378

The area of the flue in sq. ft. is then equal to $A = \frac{Q_1}{60 \times V_v}$ in which Q_1 = cu. ft. of air to be removed per hour, measured at room temperature, and V_v = velocity for the corresponding height of flue as given in Table 3, as for example, 216 ft. per min. for a flue 20 ft. high. The *U. S. Treasury Dept.* allows velocities of 300, 240 and 180 ft. per min. for 1st, 2nd, and 3rd floor vent flues respectively, in buildings not over three stories high.

If an *aspirating coil* is to be used it must be proportioned like an indirect radiator and be designed to heat all the air flowing at least 20° above the maximum outside temperature for which the flue must operate. It will be found much more economical to remove this air by means of an electric disc or propeller fan than by means of an aspirating coil and natural draft, provided electric current is available at the usual rates.

Heat and Vent Registers. The *heat registers* in this system, when 7 or 8 ft. above the floor, should be so proportioned as to make the net free area equal to the area of the warm air flue. This will require a register having 50% free area to be twice the size of the flue as, for example, an 8″ x 18″ flue will have a 16″ x 18″ register. But, if the register is near or in the floor its free area must be great enough to keep the outlet velocity down to 3 ft., or at most 4 ft. per second, to prevent disagreeable drafts and also to prevent all the heated air from rushing directly to the ceiling without having time to diffuse into the room.

The *vent registers* should always be placed at or in the floor for service during the heating season. Located in this manner they will take air that has already cooled to room temperature, and therefore no heated air will be withdrawn, and the ventilation of the room will be more economically accomplished. The size of these registers should be such that the free area will equal the vent duct area, resulting in a commercial size of twice the duct area for registers with a free area 50% of the gross area. The fact that the outflow velocity may be above 3 ft. or 4 ft. per second is of no consequence in this case as its effect is not felt in the room at all.

For *summer ventilation*, or for removing heat from a room, it is necessary to place the vent registers at or in the ceiling, and for this reason it is customary, in many cases, to install vent registers at floor and ceiling and connect then into the same vent duct. Both registers should never be open at the same time, and the top register is always kept closed so long as it is necessary to supply heat to the room.

The application of the foregoing data to a typical case is shown in the following calculations:

Example. *Indirect heating problem:* Case I.—*Given* a room with a heat loss, including leakage, of 20,000 B.t.u. per hour when the inside temperature is 70° F. and the outside temperature is 0° F. Indirect hot water radiators are to be used at a mean temperature of 170° F., with register 7′–0″ above the floor. There are no direct radiators, and there is *no ventilation requirement*, but vent flue is 30′–0″ high.

Find the quantity of air required, the size of the indirect radiator, warm-air and cold-air ducts, vent ducts and registers.

Calculations: For hot water, air will leave the radiator at about 105° F., and enter room at 100° F., losing 5° F. in the duct. The value of K, based on a velocity of 240′ per min. through the free area of the radiator, is about 2 B.t.u., (Table 8) hence, $R = K\left(t_w - \dfrac{(t_3 + t_o)}{2}\right) = 2\left(170 - \dfrac{105 + 0}{2}\right) = 235$ B.t.u. per sq. ft. per hr.

The weight of air required per hour is $W = \dfrac{H}{0.24\ (t_2 - t_1)} = \dfrac{20{,}000}{0.24\ (100 - 70)} = 2780$ lb. per hr. and $Q_2 = \dfrac{W}{d_2} = \dfrac{2780}{0.071} = 39{,}200$ cu. ft. passing through the register per hr.

The heating surface required must be sufficient to heat all the air from 0° to 105° F., or supply $H_1 = 0.24 \times W \times (t_3 - t_o) = 0.24 \times 2780 \times (105 - 0) = 70{,}000$ B.t.u. per hour. Then $S = \dfrac{H_1}{R} = \dfrac{70{,}000}{235} = 298$ sq. ft. net area of indirect radiator.

The actual radiator, if of the extended pin type, is usually over-rated 20% due to inefficiency of extended pins or fins. It is therefore customary to use 80% of the listed surface, or a 15 sq. ft. section $= 0.8 \times 15 = 12$ sq. ft. The actual radiator will contain $298/12 = 25$ sections rated nominally at 15 sq. ft. each. This should be set in two radiators of 13 sections each.

The size of warm-air duct is $A = \dfrac{Q_2}{60 \times V_a} = \dfrac{39{,}200}{60 \times 200 \times 2} = 1.63$ sq. ft., or 235 sq. in., making each duct 12″ x 20″.

The size of the cold-air duct is $0.80 \times A = 0.80 \times 235 = 188$ sq. in. or 10″ x 18.″

The size of each vent duct is, by Table 3, $A = \frac{Q_1}{60 \times V_v} = \frac{37{,}150}{60 \times 264 \times 2} = 1.17$ sq. ft. or 168 sq. in., making duct 10″ x 16″.

Example. *Indirect heating problem:* Case II.—*Given* a 2nd floor school room with 50 scholars, in which the *ventilation requirement* is 30 cu. ft. per min. for each occupant. The room temperature is 70° F., and the outside temperature is 0° F. with a resulting heat loss, including leakage, of 60,000 B.t.u. per hour. Indirect steam radiators are to be used and operated at 2-lb. gage pressure. There are no direct radiators, and the air supply registers are to be placed 7′–0″ above the 2nd floor.

Find the entering air temperature, quantity of air required, size of indirect radiators, warm air and cold air ducts, vent ducts and registers.

Calculations: It is first necessary to find whether the air for ventilation exceeds air required for heating at 120° F., which is the allowable entering temperature at register for steam.

Air for heating only is, $W = \frac{H}{0.24\,(t_2 - t_1)} = \frac{60{,}000}{0.24\,(120 - 70)} = 50{,}000$ cu. ft. per hour, but ventilation requires that $30 \times 60 \times 50 = 90{,}000$ cu. ft. per hour be supplied.

The entering air temperature then is $t_2' = \frac{H + 0.24\, W_v\, t_1}{0.24\, W_v} = \frac{60{,}000 + 0.24 \times 0.075 \times 90{,}000 \times 70}{0.24 \times 90{,}000 \times 0.075}$ $= \frac{173{,}500}{1620} = 107°$ at the register since $W_v = Q_v \times 0.075 = 6750$ lb. Since there is a 10° loss in the warm air flue to 2nd floor, the air must leave the radiator at $t_3 = (107 + 10) = 117°$ F.

Either we must increase the velocity of flow up to not exceed 5 ft. per sec. through the radiator, as shown in Table 6, until the desired final temperature is obtained, using the standard spacing for which this table was made, or we may increase the spacing of the radiator sections and free area, using a lower velocity in case we exceed 5 ft. per sec. by the first method. Reference to Table 6 shows that with 250 cu. ft. of air passing per 1 sq. ft. per hr. that with 1″ pins we have a final temperature of 118° F. and a value of $K = 3.20$.

Assuming we can get this high rate, and (in this case) velocity of flow as well, to flues supplying 2nd floor rooms, we will have for the heating surface required, since $R = K\left(t_s - \frac{(t_3 + t_o)}{2}\right) = 3.2\left(219 - \frac{118 + 0}{2}\right) = 510$ B.t.u. per hr., that $S = \frac{H_1}{R} = \frac{0.24 \times 6750 \times (118 - 0)}{510} = 376$ sq. ft.

If the free area had been increased resulting in a lower velocity for this temperature rise, the value of K and R would be less and more surface would be required than that obtained above.

The actual radiator required, using 15 ft. sections of extended pin radiators, discounted 20% as in Case I, must have $\frac{376}{12} = 31$ sections. This should be set in three radiators of 11 sections each, supplying 2 or 3 warm-air supply flues.

The size of each warm-air duct is $A = \frac{Q_2}{60 \times V_a} = \frac{90{,}000}{60 \times 330 \times 3} = 1.52$ sq. ft. or 218 sq. in., making each duct 12″ x 18″. $V_a = 330$ ft. has been taken from Table 2 by interpolation at the actual flue temperature of 110° for 2nd floor rooms. If two flues are used each duct will be 12″ x 27″ in area.

The size of cold air and vent flues may be determined, as in Case I.

The size of registers will depend on the net free area and if same velocity is allowed through the free area (taken as 50% of the gross area), as in vertical duct, the size of each register will be 18″ x 24″.

DETERMINATION OF HEAD AVAILABLE AND REQUIRED

Motive Head. The *head or pressure available* for producing flow in a gravity-indirect heating system is due to the difference in weight between the column of warm air in the flue and a column of cooler outside air (Fig. 2.) of the same height and cross-sectional area. This difference in weight is due to the fact that the warm-air column weighs less per cu. ft. than the other column,

and therefore this difference will increase directly with the height of columns for any given temperature difference.

This *difference in weight* or pressure is usually expressed in feet of air at the average temperature of the warm-air column. It may also be stated in inches of water at 70° F., and is easily found for two equal columns each 1 sq. ft. in area as follows: Consider the warm-air column (Fig. 2) extended by the amount H until its weight is just equal to the imaginary cold-air column of the same height E as the flue. Under these conditions no flow will take place, but since in practice the short column H does not exist, being constantly removed as it leaves the flue head, there is always an unbalanced head H available and equal to the weight or height of this secondary column of warm air. Equating the weights of these two columns for the condition when they are in equilibrium we have:

$$(E + H)\,d_a = E\,d_o \text{ or}$$

$$H = \frac{E\,(d_o - d_a)}{d_a},$$ and since E is always known, being the vertical distance between the center of the radiator and the register, the operating or available head can be computed for any height of flue and temperature difference. This head must overcome all resistances to flow due to friction, entry losses, and register grilles, and maintain the final velocity head at discharge.

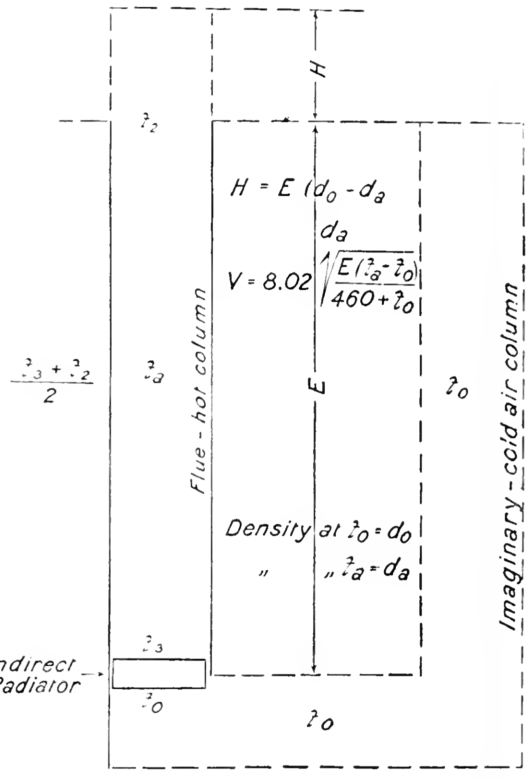


Fig. 2. Diagram Showing Head Available in Gravity-Indirect Heating.

Heads to be Overcome. It is therefore necessary that in order for a gravity-indirect system to operate that *the head available must at least be equal to the sum of all resistances, plus the final velocity head.* We may readily test any given system by finding the sum of all the heads to be overcome, which must of course be less than the head available, as above determined.

These resistance heads are.

h_1 = head lost by entry to intake of cold-air duct,

$$= 0.5\,\frac{V_o^2}{2g}$$ if there is no register. With a register this loss may become **1.5 to 2** $\frac{V_o^2}{2g}$.

h_2 = head lost by entry to base of hot-air flue from radiator,

$$= 0.5\,\frac{V_3^2}{2g}.$$

h_3 = head lost by friction in cold and hot-air ducts,

$= f \times \frac{LR}{A} \times \frac{V_a^2}{2g}$, where L = total equivalent length of the duct in feet, including all elbows, R = the perimeter in feet, A = area in sq. ft., and f = coefficient of friction, about 0.0045 for low velocities as in gravity-indirect systems. See curves for values of f in the chapter on "Hot Blast Heating." g = 32.16 ft. per sec.

h_4 = head lost through register grille

$= 1.5 \frac{V_2^2}{2g}$.

h_5 = head lost through the indirect radiator,

= approx. 0.5 ft. as determined by tests on indirect radiators one section deep at velocities of from 200 to 300 ft. per minute through the free area. The velocities V_o, V_1, V_2, V_3, and V_a are measured at temperatures t_o, t_1, t_2, t_3 and t_a respectively.

The final velocity head, $h_v = \frac{V_2^2}{2g}$ and must be added to the sum of all the other heads.

The head lost by *friction at elbows* and turns is allowed for by adding to the measured length of duct the equivalent of these fittings expressed in feet of straight pipe, as determined from Table 4 following:

TABLE 4

EQUIVALENT LENGTHS OF STRAIGHT PIPE FOR ELBOWS OF VARYING RADII, EXPRESSED IN DIAMETERS OR WIDTHS

Ratio of Inside Radius of Elbow to Pipe Diameters	Number of Diameters or Widths to be Added	
.5	29.0	NOTE.—It is advisable to add an extra elbow to allow for turn made by air in entering the indirect radiator.
1.0	10.3	
1.5	6.0	
2.0	4.4	
2.5	4.6	
3.0	4.8	

The *partial heads* may be exactly calculated, as indicated above, and then corrected for feet of air column at the temperature t_a in hot air flue, or an average constant velocity may be assumed throughout and taken as equal to velocity V_a in the vertical heat flue, at temperature t_a.

The total of all lost heads, including friction and the final velocity head, is:

$$H_x = h_1 + h_2 + h_3 + h_4 + h_5 + \frac{V^2}{2g}$$

Then, since $H_x = \frac{E\,(d_o - d_a)}{d_a} = 0.5\frac{V_o^2}{2g} + 0.5\frac{V_3^2}{2g} + f\frac{L\,R}{A}\frac{V_a^2}{2g} + 1.5\frac{V_2^2}{2g} + 0.5 + \frac{V_2^2}{2g}$

$= \left(3.5 + f\frac{LR}{A}\right)\frac{V_a^2}{2g} + 0.5$ approximately, using V_a, and allowing for 1 register.

H = head available in feet of air at temperature $\frac{t_3 + t_2}{2}$ for producing flow.

V_a = velocity in feet per sec. at temperature t_a, taken as average throughout.

d_o = density of outside air at temperature t_o, and d_a = density of heated air at temperature

$\frac{t_3 + t_2}{2} = t_a$, average in hot air flue.

Example. The application of the above method to an actual example is best shown by considering the school room problem already given in Case II under the design of a gravity-indirect heating system.

The head available must equal or exceed the sum of all t e resistance heads plus the final velocity head.

The *head available* is $H = E\dfrac{(d - d\ )}{d_a} = 20\dfrac{(0.086 - 0.070)}{0.070} = 4.57$ ft. of air at 107°.

The *resistances to be overcome* are the friction in a duct, 12″ x 18″ in area, 40 ft. in length, with three close elbows where radius equals ½ depth, and the inlet and two register losses.

By equation already given $H_x = \left(3.5 + f\dfrac{LR}{A}\right)\dfrac{V_a^2}{2g} + 0.5$, allowing for one register.

$$L = 40 + \frac{3 \times 29 \times 12}{12} = 127,\ f = 0.0045,\ R = 5,\ A = 1.5,\ V_a = 360 \div 60 = 6.0 \text{ ft. per second.}$$

$$H_x = \left(3.5 + 1.5 + 0.0046\frac{127 \times 5}{1.5}\right)\frac{6.0^2}{64.4} + 0.5 = 6.95 \times 0.56 + 0.5 = 4.38 \text{ ft. of air at } 107°.$$

The expression for H_x includes an extra 1.5 to allow for a cold-air register as well as a heat register. The head available is slightly greater than the sum of all the heads to be overcome.

ALTERNATIVE METHOD OF DESIGNING A GRAVITY-INDIRECT SYSTEM

A somewhat simpler method for finding the size of indirect radiators and ducts, limited to three assumed temperatures for the air entering the room, has been proposed by *E. T. Child.* By the use of a simple table and two sets of curves the amount of radiation and flue area are readily found when the heat loss is known.

Basis for Tabulating the Data Used. In this method the flue velocities for various floors, or heights of flue (Column 2, Table 5), are taken at one-third the theoretical (Columns 3, 4 and 5), and the radiator sections are so spaced for each final temperature as to give a *definite and constant increase in temperature* and also a *constant volume and velocity* of air flow (Columns 12 to 17) through the free area, as will be seen by reference to Table 5. This results in a fixed spacing of sections for each final temperature so that, at 100° F., a constant transmission of 400 B.t.u. takes place, at 125° the transmission is 375 B.t.u. and at 150° the transmission becomes 350 B.t.u. per sq. ft. per hr. for steam at 2 lb. gage, air entering at 0° F. Since the heat transmission is thus fixed it is possible to tabulate the sq. ft. of indirect surface required to heat the air passing through a flue 1 sq. ft. in area (Columns 6, 7, and 8), and conversely the sq. in. of flue area supplied by 1 sq. ft. of heating surface are readily determined and tabulated, (Columns 9, 10 and 11). Thus, with a flue 10 ft. high, velocity 250 ft. per min., at 125° F. air temperature, it will require 100 sq. ft. of indirect surface at an efficiency of 375 B.t.u. per sq. ft. to warm the air passing up a stack 1 sq. ft. in area when the entering air temperature is zero. Each sq. ft. of this radiation is therefore supplying 1.44 sq. in. of flue.

Use of the Tables and Curves. The application of the tables and curves is then made as follows: (1) Find the total heat loss H from the room in the customary manner. (2) Select a flue temperature of 100°, 125°, or 150°, and the corresponding rate of heat emission. (3) Find the available heat; that is, that which may be used for warming the room, per sq. ft. of radiation. This is the total B.t.u. per 1 sq. ft. given off by radiator less the heat loss in transmission plus the heat to raise the outside air to room temperature $= \dfrac{5 + 70}{\text{flue temp.}} \times R$, and $R =$ B.t.u. transmitted per 1 sq. ft. (Columns 6, 7 and 8 are based on 400, 375, 350 B.t.u. for steam and 300 for hot water.) (4) Divide the heat loss H by the available heat, $H \div$

$\left(R - \frac{75}{\text{flue temp.}} \times R\right)$ = sq. ft. of indirect radiation. (5) Select the flue size from curves using height and sq. in. of flue per 1 sq. ft. of radiator as coordinates for the curve of proper temperature and heating medium.

TABLE 5

GRAVITY-INDIRECT HEATING

(*E. T. Child*)

This table gives the square feet of indirect radiation required per 1 sq. ft. of warm air flue. Flues range from 5′ to 50′ high, and air in same taken at 100°, 125° and 150° F. See also curves—(Fig. 3).

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
	Height of flue in feet.	Velocity in Flue, Feet per Min.			Square Feet of Radiation 100° | 125° | 150°			Sq. In. of Flue per Sq. Ft. of Radiation			Volume per Sq. Ft. of Heater per Hour			Velocity of Air Over Heater Feet per Min.		
		Flue Temp.			B.t.u. per SQ. FT. per HR.			Flue Temp.								
		100°	125°	150°	400	375	350	100°	125°	150°	100°	125°	150°	100°	125°	150°
Steam	5	163	180	200	49	72	103	*2.94	2.00	1.40	200	150	117	240	180	140
	10	225	250	275	68	100	141	2.12	1.44	1.02	200	150	117	240	180	140
	20	325	360	400	98	144	206	1.47	1.00	0.70	200	150	117	240	180	140
	30	400	437	475	120	175	244	1.20	0.82	0.59	200	150	117	240	180	140
	40	450	500	550	135	200	283	1.06	0.72	0.51	200	150	117	240	180	140
	50	510	560	625	153	224	321	0.94	0.64	0.44	200	150	117	240	180	140
	Steam at 2 lb. gage = 219° F. Flue Velocity = ⅓ theoretical				B.t.u. per Sq. Ft. per Hr. = 300			$\frac{*1.44}{49}=2.94$								
Hot Water	5	163			†65			2.22			150			180		
	10	225			90			1.55			150			180		
	20	325			130			1.11			150			180		
	30	400			160			.90			150			180		
	40	450			180			.80			150			180		
	50	510			204			.70			150			180		

* Water at 180° $\dagger\frac{163 \times 60 \times .02 \times 100}{300} = 65$

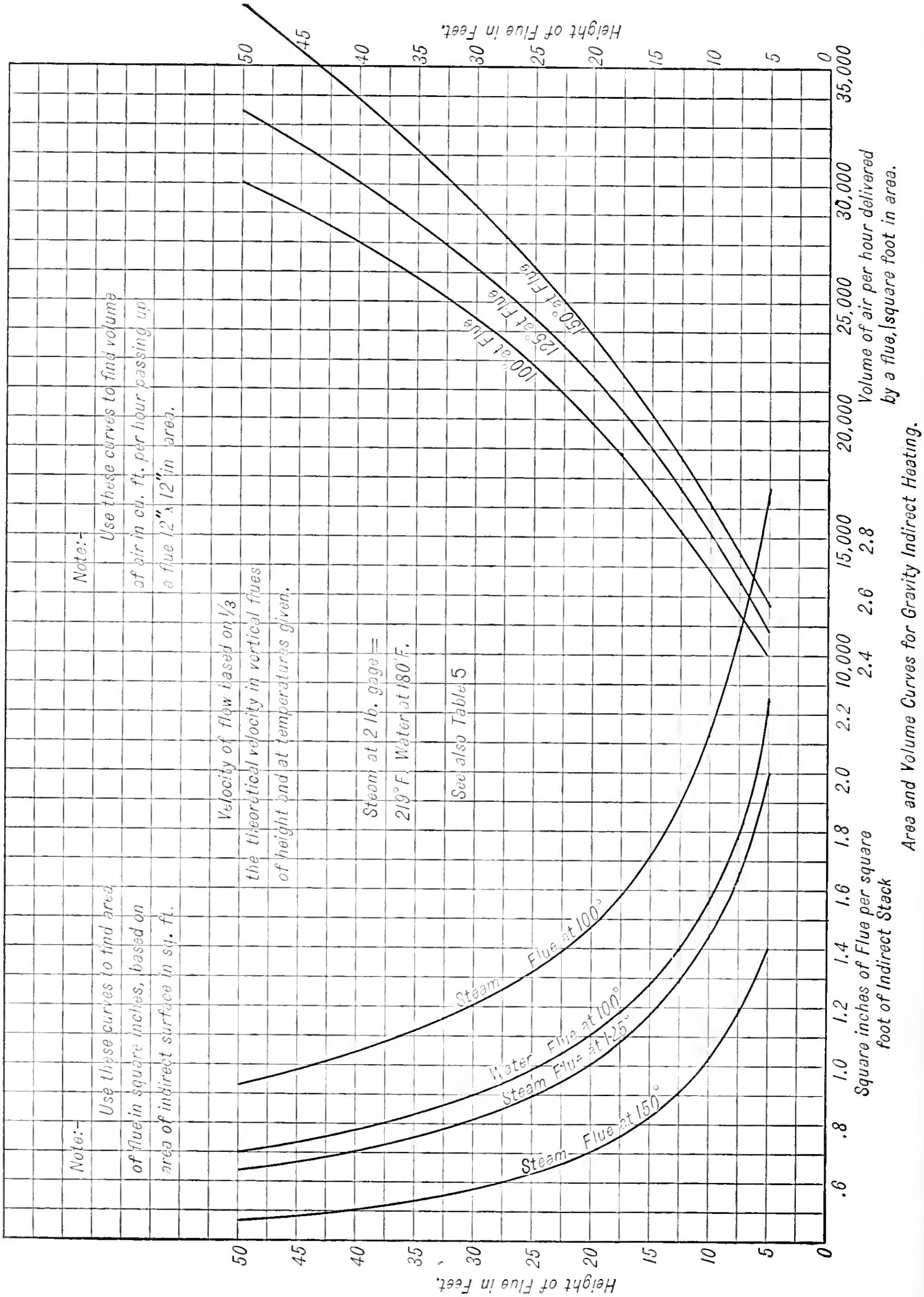
Example. A flue 20 ft. high with air leaving the radiator at 125° F. will require 1 sq. in. in the flue for each sq. ft. of indirect steam radiation.

The second set of curves makes it possible to determine the area of warm air flue if the height and flue temperature are known, as each curve is plotted to indicate the air volume which a flue 1 sq. ft. in area will handle.

Example. A flue 1 sq. ft. in area, 10 ft. high, with air at 125° will deliver 15,000 cu. ft. per hour, and if the room in question require 30,000 cu. ft. twice this area is necessary or the actual flue must be 12″ x 24″.

GRAVITY-INDIRECT RADIATOR PERFORMANCE

Heat Transmission and Temperature Rise. The heat transmission of indirect radiators is influenced by so many factors, including the type of radiator itself, that it is impossible to make practical use of the data unless certain conditions are fixed at the outset. For example, by reference to Table 6 it will be apparent that unless the rate of air flow over the radiator (Column 1) is fixed for a given steam pressure and entering air temperature it will be impossible to determine either the temperature rise (Columns 2 and 3) or the coefficient of transmission (Columns 6 and 7).



Height of Flue in Feet.
Volume of air per hour delivered by a flue, 1 square foot in area.
Area and Volume Curves for Gravity Indirect Heating.
Square inches of Flue per square foot of Indirect Stack
Note:- Use these curves to find volume of air in cu. ft. per hour passing up a flue 12" x 12" in area.
Note:- Use these curves to find area of flue in square inches, based on area of indirect surface in sq. ft.
Velocity of flow based on 1/3 the theoretical velocity in vertical flues of height and at temperatures given.
Steam at 2 lb. gage = 219° F. Water at 180° F.
See also Table 5
150° at Flue
125° at Flue
100° at Flue
Steam Flue at 100°
Water Flue at 100°
Steam Flue at 125°
Steam Flue at 150°

Fig. 3

TABLE 6

HEAT TRANSMISSION OF INDIRECT RADIATORS OF THE EXTENDED PIN-TYPE, STANDARD SPACING. STEAM

(J. R. Allen)

Cu. Ft. of Air per Hour per Square Feet of Radiation	Increase in Temperature of the Air Passing the Radiator		Pounds of Steam Condensed per Hr. per Sq. Ft. of Radiation		B.t.u. per Sq. Ft. of Radiator per 1° Difference Between Air and Steam	
	¾″ Pin	1″ Pin	¾″ Pin	1″ Pin	¾″ Pin	1″ Pin
1	2	3	4	5	6	7
50	147	140	.125	.15	.80	.95
75	143	137	.170	.21	1.17	1.27
100	140	135	.240	.26	1.51	1.60
125	138	132	.295	.31	1.85	1.90
150	135	129	.355	.36	2.22	2.20
175	132	126	.410	.41	2.57	2.47
200	130	123	.470	.45	2.90	2.72
225	127	120	.530	.49	3.25	3.00
250	123	118	.585	.53	3.60	3.20
275	121	115	.645	.57	3.90	3.40
300	119	112	.700	.61	4.22	3.60

NOTE.—Entering air at zero, and steam at 227° F., or 5 lb. gage. Use about 150 cu. ft. per hr. for residences, and 200 cu. ft. for schools. Standard pin = ¾″ and School pin = 1″. Air volume measured at 70° F.

It is also necessary to know at about what rate the air will flow over the radiator when assembled in a heating stack, and two of the conditions ordinarily met with are indicated as a guide to the designer. An increase in the rate and velocity of flow through the free area of the radiator results in a lower final temperature of the leaving air but increases the coefficient of heat transmission for the same free area and entering air temperature.

An increase in the entering air temperature results in an increase in the final temperature of the leaving air for the same velocity, and free area as shown in Table 7.

The heat transmission of indirect hot-water radiators of the extended pin type, with the hot water at an average temperature of 170°, is given in Table 8, taken from the report of the *Committee on Hot-Water Heating, A. S. H. and V. E.*

Vento Sections for Gravity Systems. The performance of Vento steam and hot-water indirect radiation, operating under gravity circulation, is shown in Figs. 9, 10 and 11 from tests by the *American Radiator Co.* See Table 11 for dimensions.

TABLE 7

FINAL TEMPERATURES WITH INDIRECT RADIATORS OF THE EXTENDED PIN-TYPE. AIR ENTERING ABOVE ZERO

(J. R. Allen)

Temperature of Entering Air in Degrees F.	Temperature of Air Leaving Radiator with a Velocity Equivalent to the Cubic Feet of Air Given Below per 1 Square Foot Radiation			
	200 Cu. Ft.		150 Cu. Ft.	
	¾″ Pin	1″ Pin	¾″ Pin	1″ Pin
1	2	3	4	5
0°	130	125	135	128
10°	134	128	139	132
20°	139	132	144	136
30°	144	136	149	140
40°	148	141	153	144
50°	153	144	158	146

NOTE.—This table for initial temperatures other than zero, steam at 227° or 5 lb. gage. Volume of air passing is in cu. ft. per hour at 70° F.

TABLE 8

HEAT EMISSION OF GRAVITY-INDIRECT HOT WATER RADIATORS. INITIAL AIR TEMPERATURES = 0° F.

Velocity of Air in Feet per Min. Through Free Area	B.t.u. per Sq. Ft. per Hr. per 1° Difference in Temp.
174	1.70
246*	2.00
300	2.22
342	2.38
378	2.52
400	2.60
428	2.67
450	2.72
474	2.76
492	2.80

* NOTE.—At 246 ft. velocity the final air temperature = 110° approximately, hence each sq. ft. of indirect radiation gives off $R = \left(170 - \frac{110 + 0}{2}\right) \times 2.00 = 230$ B.t.u. per hour.

Piping Connections for Indirect Radiators. The *pipe connections* to indirect radiators (Figs. 13 and 14) are always made with separate inlet or supply and separate return. These connections may be at the same end of the radiator, and should always be separately valved with combination screwed unions between valves and radiator to provide for disconnection. A suitable air-tight shield or plate must be provided at the points where pipes pass through the casing wall whether the latter is of metal or brick. An automatic *air valve* should be installed at the return elbow (Fig. 13).

The *elevation of the return tapping* on the indirect radiator must be at least 24 inches above the boiler water line on all gravity systems (Fig. 13). If necessary to accomplish this, the boiler must go in a pit of proper depth.

The *size of flow and return pipes,* for either steam or water radiation may be determined by expressing the indirect surface in terms of direct surface and then use the pipe size required for two-pipe gravity work. Thus for 500 sq. ft. of indirect surface, $R = 350$ B.t.u. per sq. ft. per hour $= 500 \times \frac{350}{250} = 700$ sq. ft. direct steam radiation for which the pipe size may be determined. See the chapter on "Direct Steam Heating."

DIMENSION DATA FOR GRAVITY-INDIRECT RADIATORS

Gravity-indirect radiators are made in a great variety of forms and sizes and usually consist of cored cast iron *sections* with extended surfaces which may be built up into *stacks* of the required total area by the use of screw or push nipples, or through-bolts in case flanged sections are used.

In the case of extended pin radiators or extended flange radiators (Figs. 4 and 6) these solid extensions are not as efficient as the *prime* surface, such as the hollow projections on the Vento sections (Fig. 8). It is therefore the practice of many designers to *discount the measured surface* of such radiators as have solid extensions by approximately 20%. However, if the transmission coefficient has been determined, using the actual measured surface, then this surface may be used in selecting the number of sections required, and the full rating allowed.

Extended Pin Indirect Radiators. The extended pin radiators are made with two lengths of pins; in one the pins are ¾″ and in the other 1″ long. The *H. B. Smith Co.* makes both the *Gold Pin* and the *School Pin* indirect radiators for steam and water using right and left screw nipples 2″ in diameter for connecting the sections together.

TABLE 9

DIMENSIONS OF R. & L. NIPPLE GOLD PIN AND SCHOOL PIN INDIRECT RADIATORS

(See Fig. 4)

	Commercial Designation	Square Feet per Section	A = Distance c to c	B = Distance Between Ends of Pins	C = Length of Pin	D or G = Total Height of Section	E = Length of Section	F = Net Height of Section	Supply and Return Tappings	Air Valve Tapping	Size R. & L. Nipples	Weight per Section, Lbs.	Free Area	
													Square Feet	Square Inches
1	2	3	4	5	6	7	8	9	10	11	12	13		
			Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	In.			
1	Gold Pin*	12	3¼	¼	¾	9	36	8½	1¼	⅜	2	62	.25	36
2	Gold Pin	15	3¼	¼	¾	11½	36	10	1½	⅜	2	77	.25	36
3	Gold Pin	20	3¼	¼	¾	15½	36	14	1½	⅜	2	106	.25	36
4	School Pin	15	4	½	1	11½	36	10	2	⅜	2	82	.42	61
5	School Pin	20	4	½	1	15½	36	14	2	⅜	2	115	.42	61

*NOTE.—The 12-foot Gold Pin radiator is made for steam only, with inlets and outlets as shown, with nipples at bottom only.

By the use of longer or shorter nipples, the "free area" may be varied as desired. Nipples vary in length by ¼" intervals.

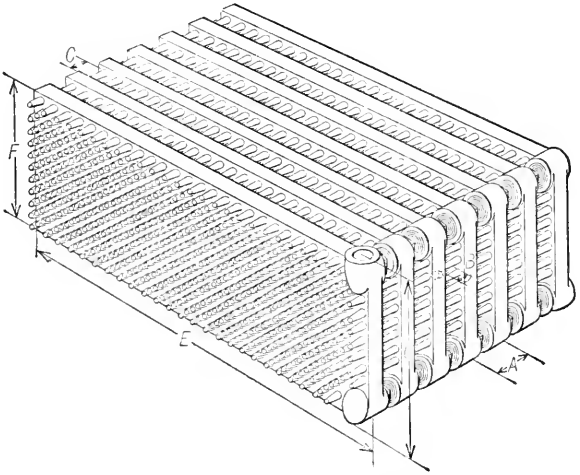


FIG. 4. RIGHT AND LEFT NIPPLE SCHOOL PIN INDIRECT RADIATOR. (STEAM OR WATER.)
(See Table 9)

A form of extended pin indirect radiator, in more or less common use, also made by the *H. B. Smith Co.*, is known as the *Drum Pin* (Fig. 5), and is built with separate flow and return headers, with nipple connections between the headers or drums and each individual section, which are entirely independent of each other. This gives an unobstructed free area for air flow between the sections as there are no connecting nipples.

Extended Flange Radiators. The extended flange radiators are made in a great variety of forms, of which the *Sterling*, made by the *American Radiator Co.*, is an example.

Heating surface of single section is 20 sq. ft.; length is 36¾"; height, 15¾". Width, each section in stack, 3½". Extra long nipples can be used to make center to center of sections in stack, 3¾, 4, 4¼ or 4½".

TABLE 10

DIMENSIONS OF DRUM PIN INDIRECT RADIATOR

(See Fig. 5)

Commercial Designation	Square Feet per Section	A = Distance c to c*	B = Distance Between Ends of Pins	C = Length of Pin	D = Height of Section at Center	E = Length of Section	F = Height of Section at End	G = Height Over Drums	H = Distance from Section to Top of Drums	K = Outside Diameter of Drums	Supply and Return Tappings	Air Valve Tapping	Size R. & L. Nipples	Weight per Section, Lbs.
6″ Drum Pin..............	11	$3\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{4}$	$8\frac{1}{2}$	$40\frac{1}{2}$	$6\frac{1}{2}$	$21\frac{1}{4}$	$6\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{8}$	1	68
10″ Drum Pin.............	16	$3\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{4}$	$11\frac{3}{4}$	$40\frac{1}{2}$	10	$24\frac{3}{4}$	$6\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{1}{2}$	$\frac{3}{8}$	1	109

*NOTE.—The distance A may be varied to give any free area desired and the length of drums made to correspond.

The "free area" with $3\frac{1}{2}''$ c. to c. is 43 inches, and the supply and return tappings are each $2''$ with $\frac{3}{8}''$ tapping for air valve. Special supply and return tappings may be furnished at

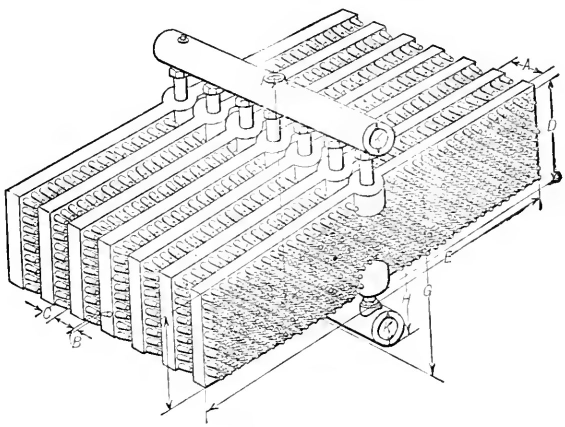


FIG. 5. DRUM PIN INDIRECT RADIATOR.

(See Table 10)

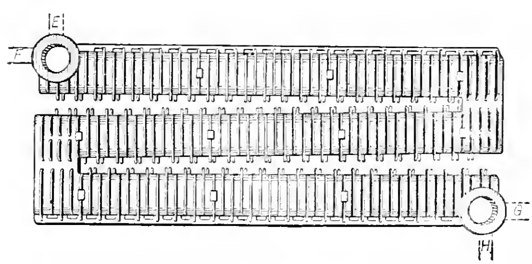
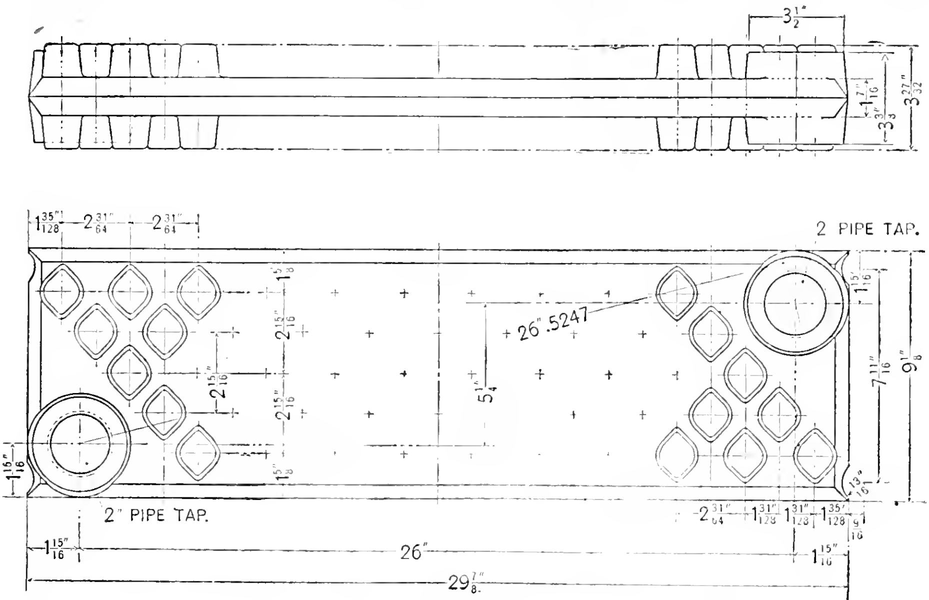


FIG. 6. STERLING INDIRECT RADIATORS FOR STEAM OR WATER.

E, F, G and *H* to meet special conditions. All sections are assembled with right and left screw nipples, the length of which may be varied, as indicated above, to give any desired spacing and corresponding "free area" between sections.

These sections provide for a positive flow of steam or water from the flow to the return side, and are especially designed to provide for any irregularity in expansion or contraction of the sections or the stack.

Extended Surface Radiators. An extended surface radiator in which all the surface is *prime*



New 30-inch Top and Bottom Connected VENTO Hot Blast Heater

FIGS. 7 AND 8.

heating surface is known as the *Vento* indirect radiator (Figs. 7 and 8), made by the *American Radiator Co.*, the performance curves for which are shown in Figs. 9, 10 and 11.

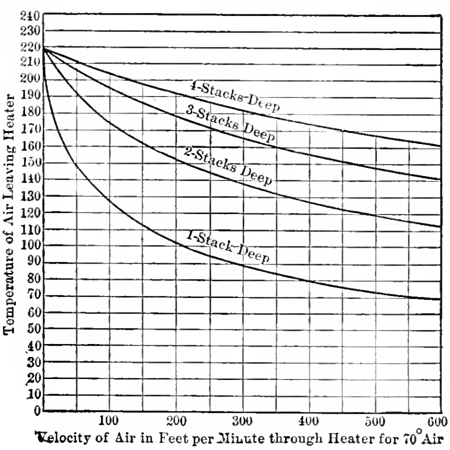


FIG. 9. FINAL TEMPERATURES WITH VENTO INDIRECT RADIATORS—STEAM.

NOTE—Steam pressure, 2 lb. gage, 219° F. Air entering at 0° F. Staggering of stacks not necessary with 4" centers.

These radiators are built up of cast iron sections spaced 4″ on centers connected with 2½″ right and left cast iron screw nipples. A somewhat similar radiator is now made by the *H. B. Smith Co.*

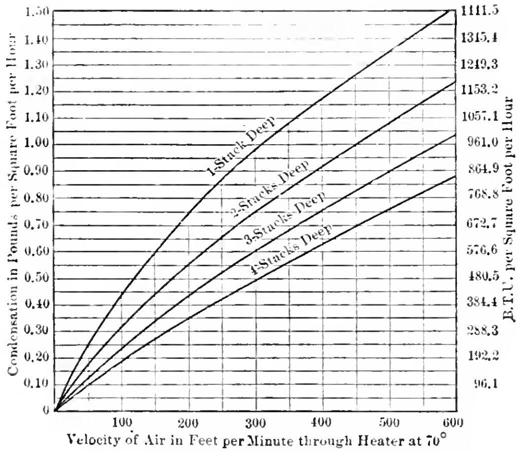


FIG. 10. HEAT TRANSMISSION AND CONDENSATION FOR VENTO INDIRECT RADIATORS—STEAM. NOTE—Steam pressure, 2 lb. gage, 219° F. Air entering at 0° F. Staggering not necessary for 4″ centers.

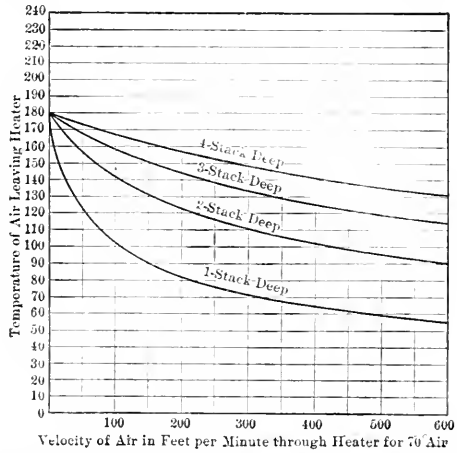


FIG. 11. FINAL TEMPERATURES WITH VENTO INDIRECT RADIATORS—HOT WATER. NOTE.—Mean water temperature, 180° F. Air entering at 0° F.

TABLE 11

DIMENSIONS OF VENTO SECTIONS

Size	Area, Sq. Ft.	Height, Inches	Width, Inches	Free Area, Sq. Ft.
30″ Section	8.00	29 7/8	9 1/8	.256
40″ Section	10.75	41 1/64	9 1/8	.350
50″ Section	13.50	50 29/32	9 1/8	.428
60″ Section	16.00	60 11/16	9 1/8	.511

NOTE.—Built for steam or water with flow and return tappings, spaced 4″ on center for gravity air circulation.

CASINGS AND DUCTS FOR INDIRECT RADIATORS

The casings and ducts for indirect radiators are usually made of sheet metal (Figs. 1 and 12) although brick chambers and brick flues (Figs. 13 and 14) are also used in large installations such as in school buildings.

Casings for Indirect Radiators. Ordinarily, galvanized sheet iron of No. 24 U. S. S. gage is satisfactory for casings, which must be made with air tight riveted or slip joints and may be

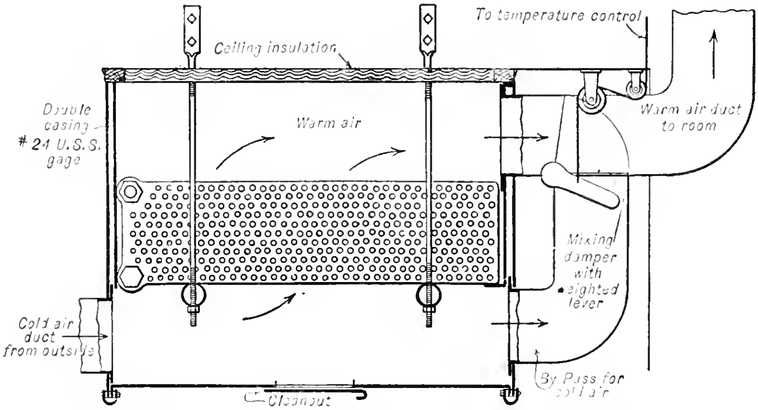


Fig. 12. Double Walled Indirect Casing with By-Pass.

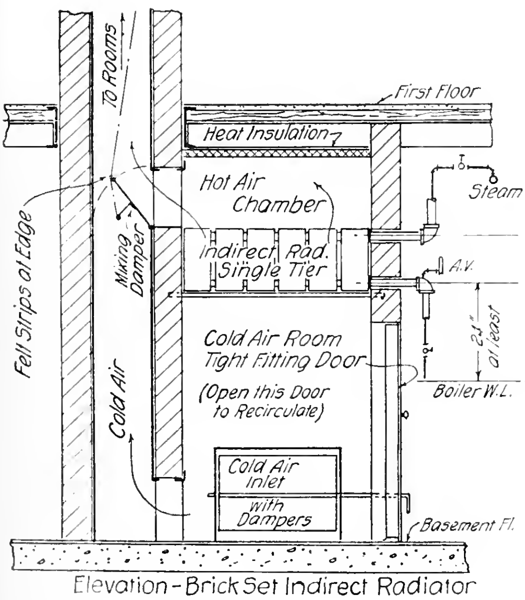


Fig. 13.

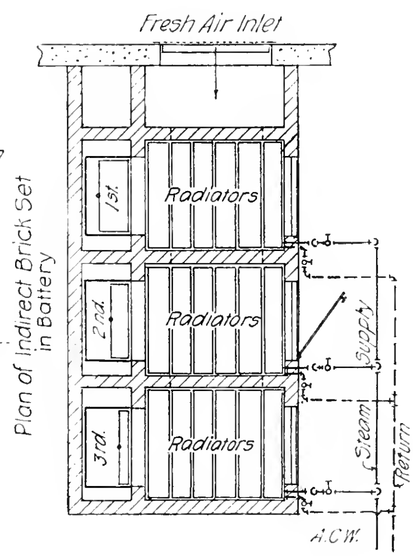


Fig. 14.

Note.—Piping connections for these radiators are described on page 303.

either single (Fig. 1) or double walled (Fig. 12). Ample insulation should be provided for the single walled casings, and the ceiling directly over all indirect stacks should be protected by at least 1″ of asbestos air-cell covering or its equivalent to prevent undue heating of the floor above.

Suitable *by-pass ducts and dampers* must be provided at each casing, as shown, so that a mixture of properly tempered air may be supplied the room in mild weather. The casing must fit the radiator closely with proper air stops, so that no cold air can by-pass the heating surfaces except as provided for at the by-pass damper. These stops are clearly shown in Fig. 12, where the lower edge of the inner casing is flanged over close against the radiator.

The horizontal *dimensions of the casing* will be determined by the number of sections in the radiator and the length of section used. The vertical dimension must include a 6″ to 8″ *cold air chamber* below the radiator and an 8″ to 10″ *hot air chamber* above the radiator. When stacks are set in two tiers about 4″ is allowed between them, and they are not staggered.

Suitable tight fitting *cleanouts* should be provided in the bottom of all indirect casings, for removing dirt that may collect in same. These are especially necessary where floor registers are installed.

Hangers of substantial form, made of 3/8″ to 5/8″ steel rods, are used for supporting indirect stacks resting on 1″ or 1¼″ pipe (Fig. 1–b); or small angles (Fig. 1–c) 1″ to 2½″ or T-irons may be used. Flat bars, 2½″ x ½″ may also serve as bearing bars. The character of the floor construction will determine the manner of fastening the head of the hanger.

Ducts and Registers. The *cold air ducts* may be run separately from an outside connection or may be taken from a *trunk duct* (Fig. 1–a) supplying several stacks, and suitable *dampers* placed at each branch connection.

Recirculating ducts may also be used (Figs. 1 and 1–a) for recirculating the air from the room, and thereby reduce the amount of heat required from the radiator, and decrease the time required for warming up an apartment when outside air is not required for ventilation. In the case of school buildings recirculation should always be practiced when the building is unoccupied, for the sake of economy.

Registers should be as plain as possible, with large free areas, and when placed in the floor must have the necessary structural strength to support the people passing over them. Tables of commercial register sizes will be found in the chapter on "Furnace Heating."

TYPICAL INSTALLATION

The *basement plan* of a typical gravity-indirect hot water heating system installed in a large residence is shown in Fig. 15. The boiler plant consists of two Walker boilers, each of which has a firebox 32″ wide by 54″ long.

The indirect *stacks* are located on the basement ceiling and are hung by rods and bars from the first floor. The indirect radiators consist of sections 15 sq. ft. in area, bolted together by short bolts. The casings are of No. 24 U. S. S. gage galvanized iron with a 12″ air space both above and below the radiator. In some cases, as shown, as many as 6 hot air pipes have been taken from one casing. There are 15 indirect stacks, containing a total of 4,920 sq. ft. of heating surface.

The *cold air ducts* are run from trunk ducts, as shown, and have an area of 18 sq. in. for each 15 sq. ft. section of indirect radiation installed.

The supply *mains* run at the basement ceiling and all radiator connections are valved at flow and return. All piping is covered with sectional pipe covering. An expansion tank 15″ in diameter by 30″ in length is connected to the high point of the basement mains just above the boiler, and vents the entire piping system and radiators, so that no air valves are required on the radiators.

DIRECT-INDIRECT HEATING

Direct-Indirect Radiators. The use of direct-indirect radiators (Fig. 16) provides for the circulation of either inside or outside air through a radiator of the *flue type*, which is placed directly

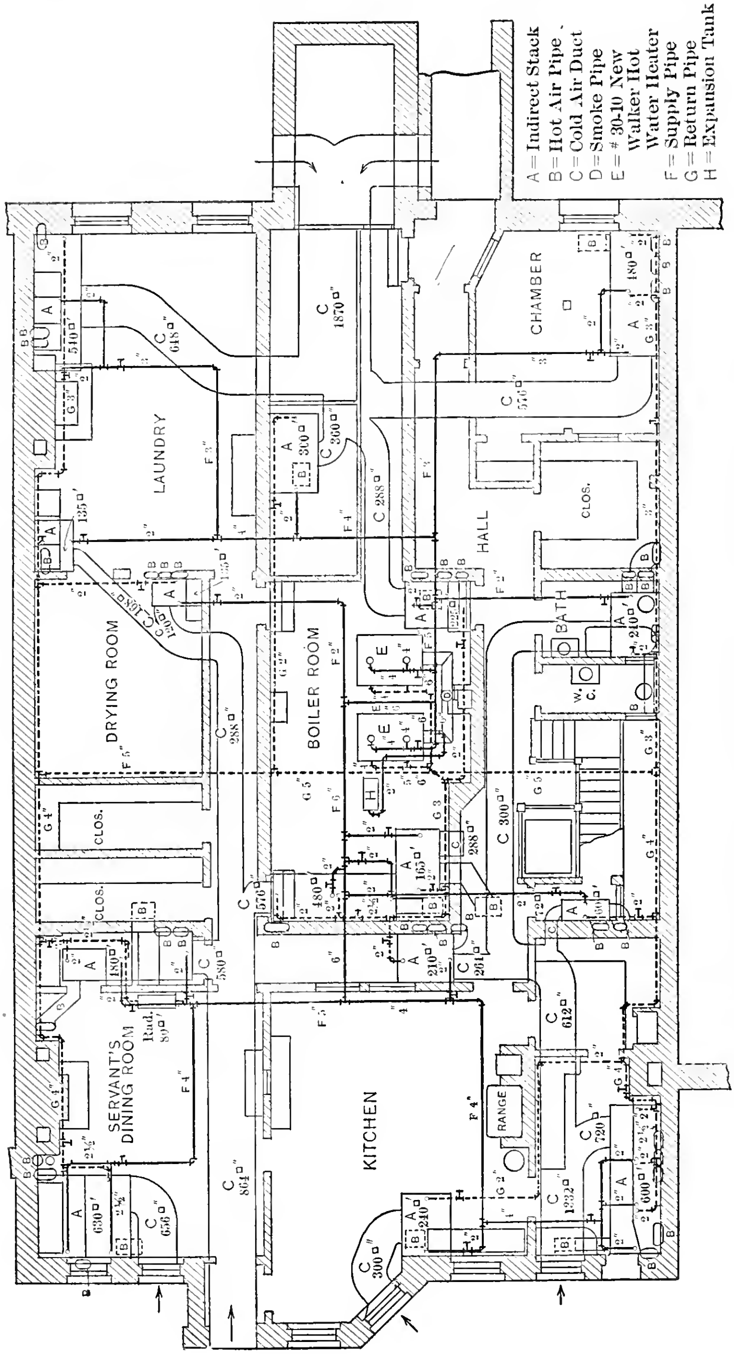


FIG. 15. Indirect Hot Water Heating System in a City Residence.—Basement Plan. (*Engineering Review.*)

in the apartment to be heated. The exterior or exposed surface of such a radiator gives off its heat in the same manner as a direct radiator, while the surface between the sections, which is formed into vertical flues, gives off its heat by convection only, like an indirect radiator.

Connections to Radiator. These radiators must be connected to the outside air by a suitable *cold-air duct*, usually run in the outside wall to connect with a cast-iron louvered *wall-box* set just under a window sill where it will be as inconspicuous as possible. It is usually necessary to place these radiators under windows. The manner of connecting this duct to the *box-base* in which the radiator sets is a most important detail of the installation, as some provision for adjustment must be made, and at the same time as tight a joint as possible must be formed. The method used by the *U. S. War and Treasury Depts.* for making this connection is clearly indicated in Fig. 16.

The box-base is furnished with two dampers operated by cam and lever, and the cam is so designed that *both dampers* can *never be open at the same time* in order to prevent the possibility of cold air blowing through into the room without passing over the radiator.

The *ventilating effect* of these radiators is rather limited, as the velocities through the cold air duct seldom exceed 5 ft. per second. In general it will require from 25 to 30 sq. feet of direct-indirect radiation to furnish air for one person.

Performance of Direct-Indirect Radiators. The *heat transmitting efficiency* of these radiators is about the same as direct radiators, when recirculating the air, as indicated in the following test reported by *J. R. Allen*. The radiator tested had 42 sq. ft. of rated surface, and was operated with steam at 212° and surrounding air at 70° F. The recirculated air was raised from 70° to 152° and left the flues with a velocity of about 171 ft. per min. The steam condensed per sq. ft. per hour weighed 0.227 lb. and the heat transmission was 1.57 B.t.u. per 1° difference. The heat distribution was divided as follows: 45% transmitted by flues and 55% radiated. The data for this radiator when taking outside air at 0° F. and operating as a true indirect radiator are not given, but from experience in the use of this type of radiation it is customary to assume an *increase of at least* 25% in the heat transmission in determining the load on the boiler, due to direct-indirect radiation.

Furthermore, in *proportioning this radiation*, based on the calculated heat loss, it is customary to arbitrarily add from 20% to 25% to the amount of direct radiation that would ordinarily be installed to offset the heat loss. Probably a more rational method is to find the heat loss from the room as above, and then add to it the heat required for raising the temperature of the air introduced through the flues. This total loss is then divided by 300 B.t.u., the heat transmission per sq. ft. of an indirect steam radiator operating with outside air. For hot water use 200 B.t.u. per sq. ft. For very cold climates, where temperatures remain below zero, hot water direct-indirect radiators should not be used, as they are liable to freeze.

Specification for Direct-Indirect Radiation. The following specification for direct-indirect radiators and ducts applies to the arrangement shown in Fig. 16 and is in accordance with the practice of the *U. S. War Dept.* for heating and ventilating barracks and army hospitals.

Direct-Indirect Radiation. The direct-indirect radiation is to be of cast-iron flue sections put together with cast or malleable iron screw or slip nipples, placed where shown on plan and of required height. Each direct-indirect radiator to be provided with cast-iron base and dampers. The box-base is to have dampers for the circulation of both inside and outside air through radiator, so arranged that either damper may be opened at will and that one must close tight before the other will open. All box-bases must have a detachable front held in place by flathead machine screws which must be easily removable with screw driver. All bases must be true and even, out of warp and operate to the satisfaction of the superintendent before and after installation. The length of box-bases to correspond with the number of sections in radiators up to twelve sections. Twelve-section box-bases to be used for radiators of 12 sections or any even number of sections larger, and 11-section box-bases for radiators of 11 sections, or any odd number of sections larger. Each direct-indirect radiator will have a cast-iron wall-box, 5 inches by 17½ inches in size, with angle slats and inside copper-wire screen. All wall-boxes must be properly and

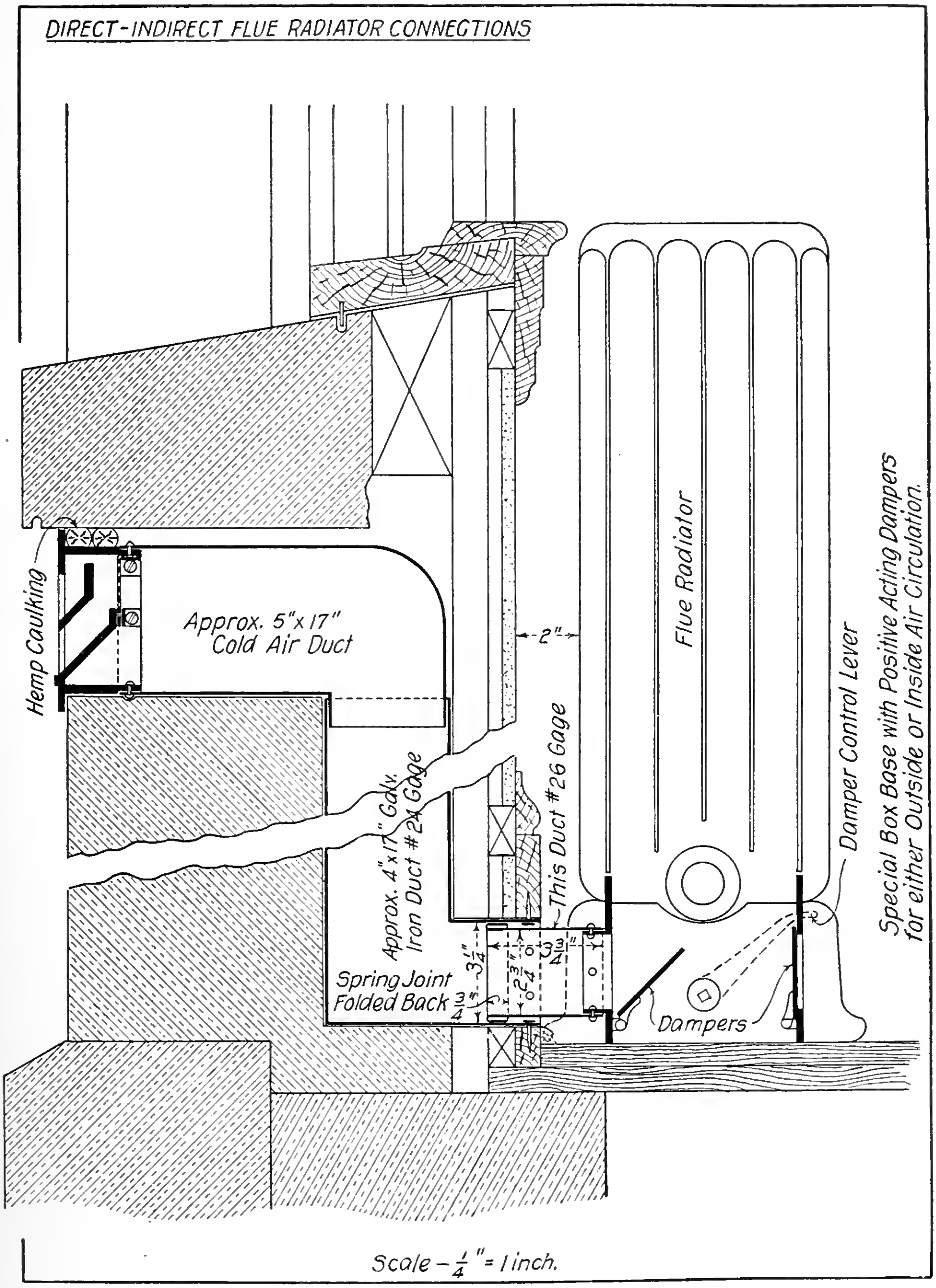


FIG. 16.

securely set in wall at elevation shown. The wall-box and box base of radiators must be tightly connected with No. 24 galvanized-iron cold-air duct; duct to be run as shown in detail drawing of same.

CHAPTER XIV

WARM-AIR FURNACE HEATING

ESSENTIAL FEATURES OF THE SYSTEM

The method of warming or heating a building by what is generally known as a warm-air furnace is termed *furnace heating*.

The Furnace and Its Location. The furnace consists briefly of a cast-iron or steel heater, containing a *combustion chamber, fire-pot* and *grate*. The heater is usually set in or encased by

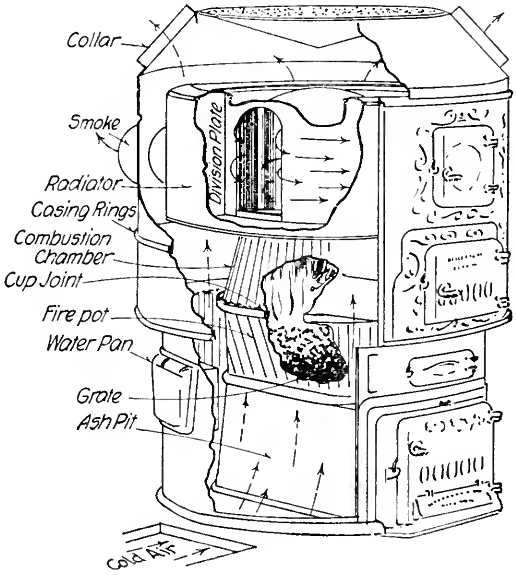


Fig. 1.

a double wall galvanized sheet steel *jacket* (Fig. 1 and 3), although brick is sometimes used instead of the steel jacket for this purpose. (Fig. 2.)

Furnaces for soft coal (Fig. 3) are usually designed with a secondary air supply or *over-draft* for admitting heated air just at the surface of the fire (Fig. 3*a*) in order to produce a more perfect combustion of the volatile combustible gases which are liberated from this fuel immediately after firing. This over-draft should be under positive control so that it may be checked or closed after the fuel has been coked.

Soft coal may also be burned efficiently in the underfeed type of furnace (Fig. 4), in which coal is fed from below by means of a plunger operating in a feed chute discharging through the center of the grate.

The furnace should be located in the basement in an approximately central position with reference to the rooms to be heated, and preferably toward the side or sides from which the prevailing winds blow in the winter time. This arrangement not only favors the more exposed

rooms on the floors above by shortening the leaders to these rooms, but also makes it possible to reduce the length of the cold-air duct, which should always be run from the exposed side of the building to the cold-air pit below the furnace. (Figs. 1, 23 and 24.)

In operation cold air is drawn from the outside through the *cold-air duct* and passed through the space between the heater and its jacket and is warmed by coming in contact with

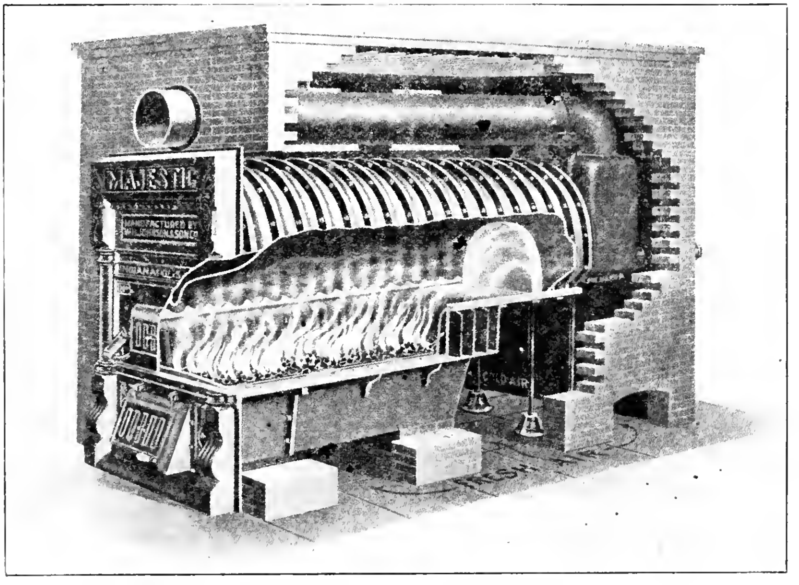


Fig. 2. Johnson Brick-Set Furnace.

the outside heated surface of the combustion chamber and radiator, which is usually just above the combustion chamber. It is then discharged through flues connected at the top of the *jacket*, furnace *cap* or *bonnet* to the rooms to be warmed.

Leaders and Stacks. These connecting flues are in reality made up of two sections, (1) the nearly horizontal round pipes in the basement known as *leaders* (Figs. 29 and 30), which connect to the *collars* on the top or conical sides of the bonnet, and (2) the vertical rectangular pipes called *stacks* (Figs. 5 and 6), which connect the *boot* (Figs. 7 and 8) at the outer end of the leader, with the double walled *register box* (Figs. 9, 20 and 21), into which the *register grille*, covering the opening into the room, is fitted. The leaders should have an upward pitch toward the base of stack of at least 1″ per 1′–0″, and for best results should not be over 12 to 15 feet in length. The boots are made in a great variety of shapes to suit actual conditions, and are simply adapters for changing from the round leader to the rectangular stack.

The stacks are usually run between the studding of interior walls or partitions (Figs. 5 and 6), since if they are placed in outside walls the cooling effect reduces their efficiency not only in temperature of air, but also in velocity of flow.

The *metal* used for leaders and stacks is usually bright IX tin, although for leaders larger than 12″, galvanized steel is employed of No. 26 U. S. S. gage, as used in ducts for "Hot Blast Heating."

The covering of all leaders, boots and stacks, as well as the furnace itself, is most important, and either a heavy grade of asbestos paper is pasted on the outside, or, as in the case of leaders and the furnace itself, asbestos air cell covering about ¼-inch thick may be used and secured

with brass bands or wire. Since the stacks must run, generally, in a 4-inch studding space, with net depth of about 3¾ inches, every effort must be made to keep them as deep as possible, and

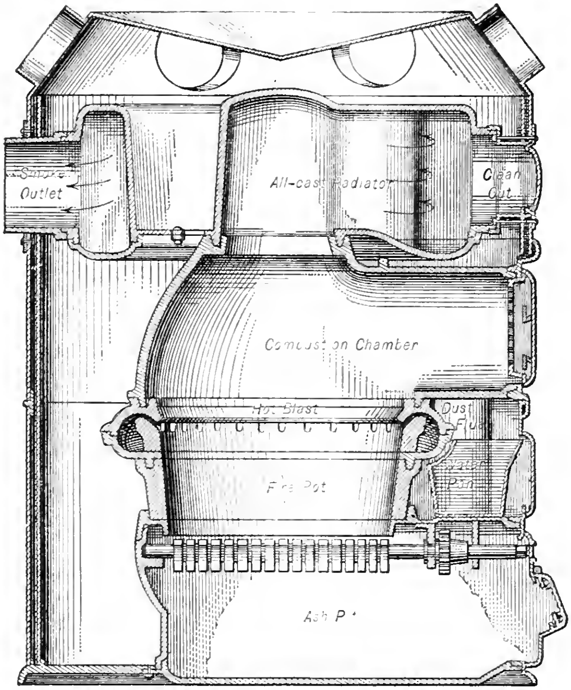


Fig. 3. Stewart Soft Coal Furnace.
(Fuller Warren Co.)

TABLE 1

GENERAL DIMENSIONS OF SOFT COAL FURNACES

Nos.	Heating Capacity: For Churches, Halls, Stores, Etc. Cubic Feet	Heating Capacity: For Residence Work Divided Into Rooms Cubic Feet	Inside Diameter of Fire-Pot Inches	Diameter of Casing Inches	Height of Castings Inches	Diameter of Grates Inches	Size of Smoke-Pipe Inches
36	16 to 20,000	12 to 14,000	20	39	45	18½	8
46	20 to 25,000	14 to 18,000	22	42	48	20½	8
56	25 to 32,000	18 to 24,000	24	46	52	22½	9
66	32 to 42,000	24 to 32,000	26½	51	57	25	9

Note.—The hot blast is in the form of a hollow ring with two air inlets located at opposite sides. This ring forms the upper part of the fire-pot wall and is just below the combustion chamber. It is joined to sections above and below by cement-filled cup joints. The part of the ring exposed to the fire has graduated openings around its upper edge. Air admitted at the inlets circulates through the ring, and after becoming thoroughly heated passes through the holes downward over the fire. The heated oxygen ignites with the gases and the smoke of the soft coal. The result of this process is an intense heat. The upper and lower halves of the hot blast ring are made of two separate castings and are joined by means of heavy lugs and bolts. This construction, found only in the Stewart furnace, insures against cracking, which is likely to happen to the one-piece hot blast contrivance.

steel lathing or expanded metal should be used in front of all such stacks, which ordinarily have only a single layer of asbestos paper covering. A more effective insulation may be provided

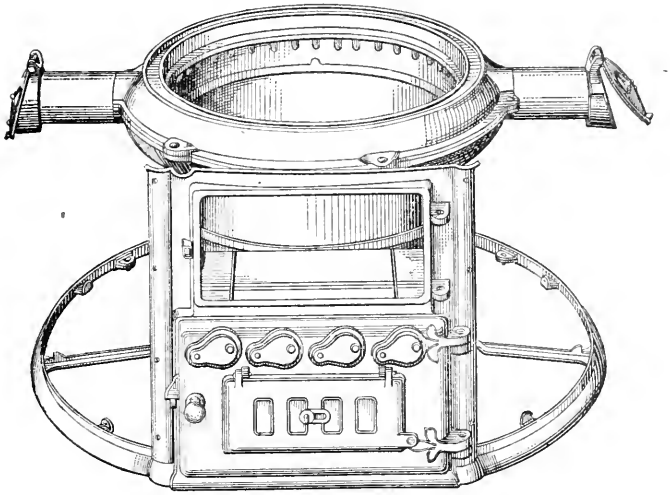


FIG. 3a. HOT BLAST SECTION OF FURNACE.

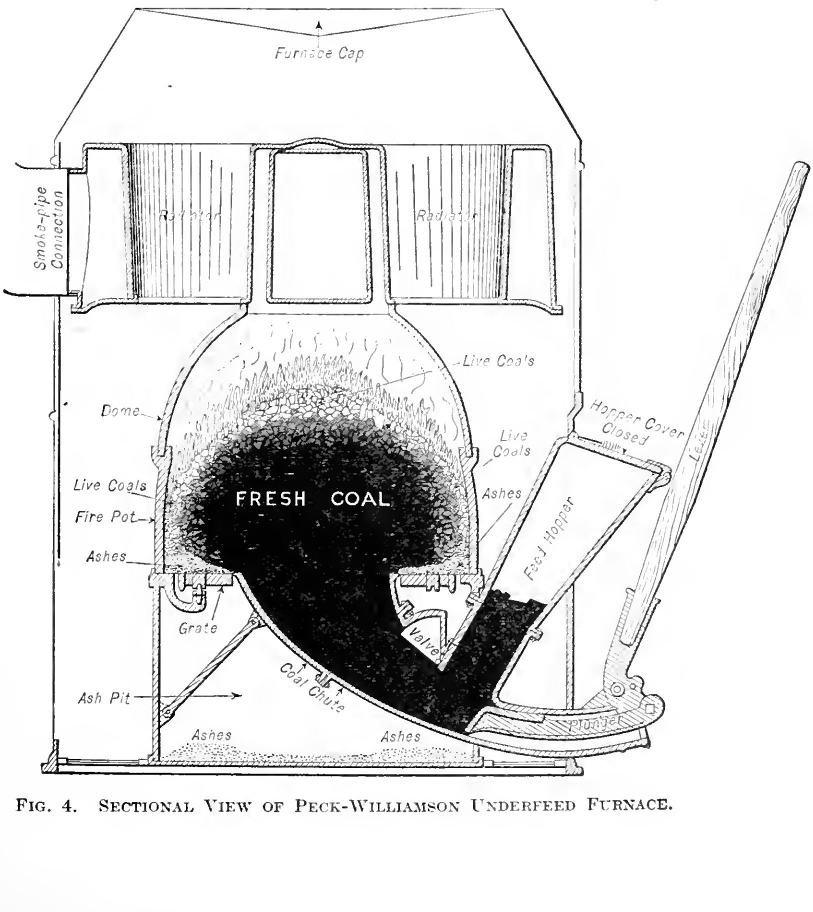


FIG. 4. SECTIONAL VIEW OF PECK-WILLIAMSON UNDERFEED FURNACE.

by using a *double wall stack* (Figs. 10, 11 and 12), in which there is an air space between the inside and outside pipes, and no asbestos covering is used. See Table 6 of this equipment, as made by the *Excelsior Steel Furnace Co.*

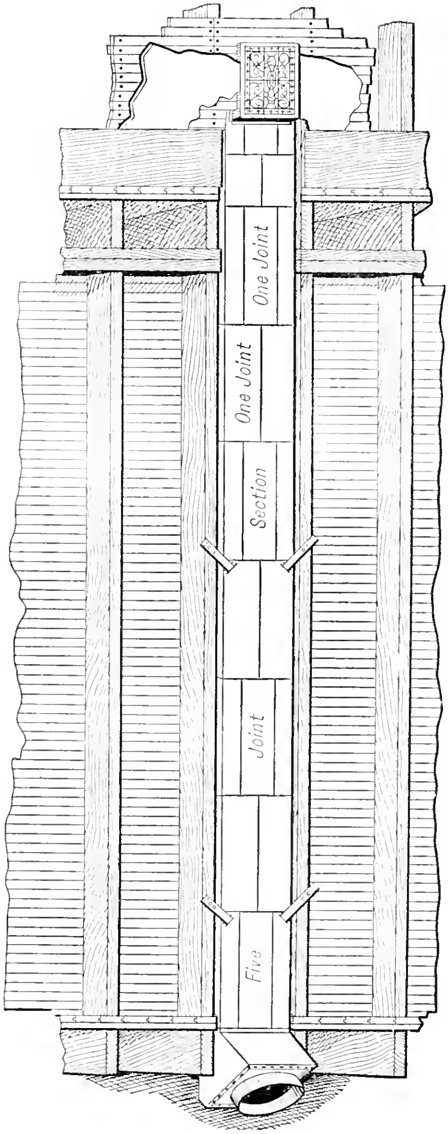


FIG. 5.
VERTICAL STACK WITH SIDE WALL REGISTER.

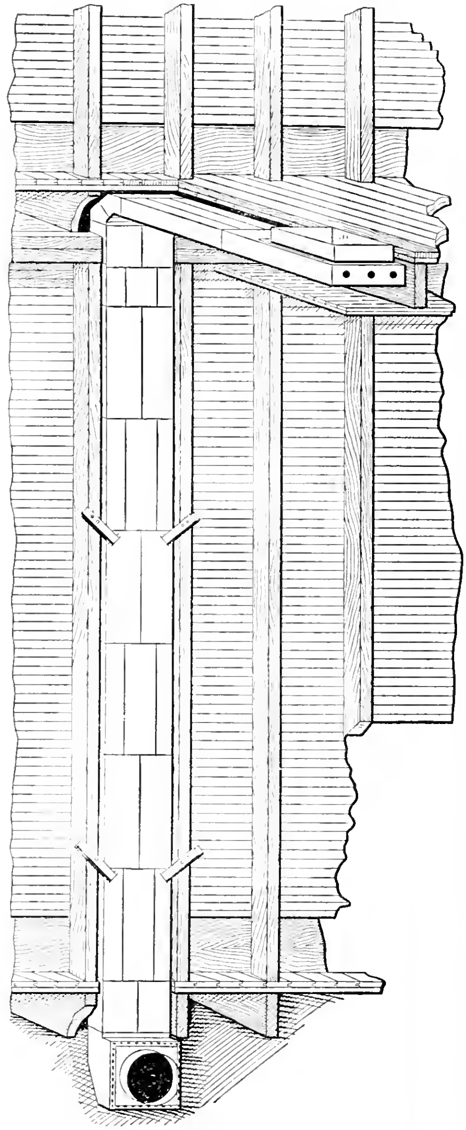


FIG. 6.
VERTICAL STACK WITH REGISTER IN UPPER FLOOR.

THE DESIGN OF A FURNACE HEATING SYSTEM

Heat Loss and Air Required. The determination of the size of the furnace, and the connecting leaders, stacks, registers, ducts, etc., is based on: (1) The actual heat loss from each

room in the building, including wall and glass transmission losses, as well as loss due to infiltration (see chapter on "Heat Transmission of Building Materials"); and (2) the amount of air to be circulated per hour, which in turn is based on this heat loss.

A building is warmed or heated by hot air by introducing the air into the rooms at a higher

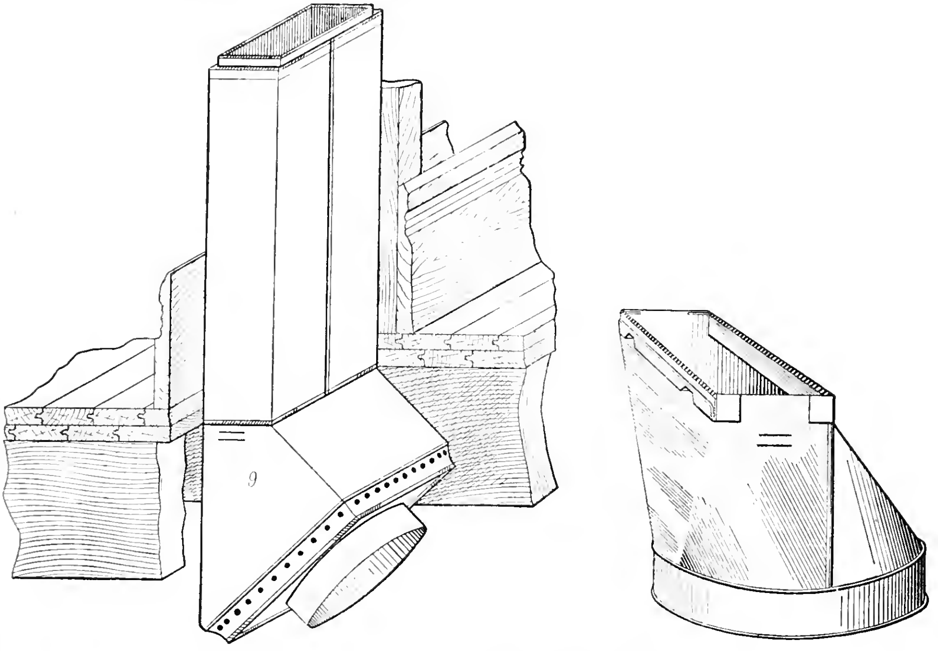


FIG. 7. STACK SECTION AND BOOT DOUBLE WALL PIPE.

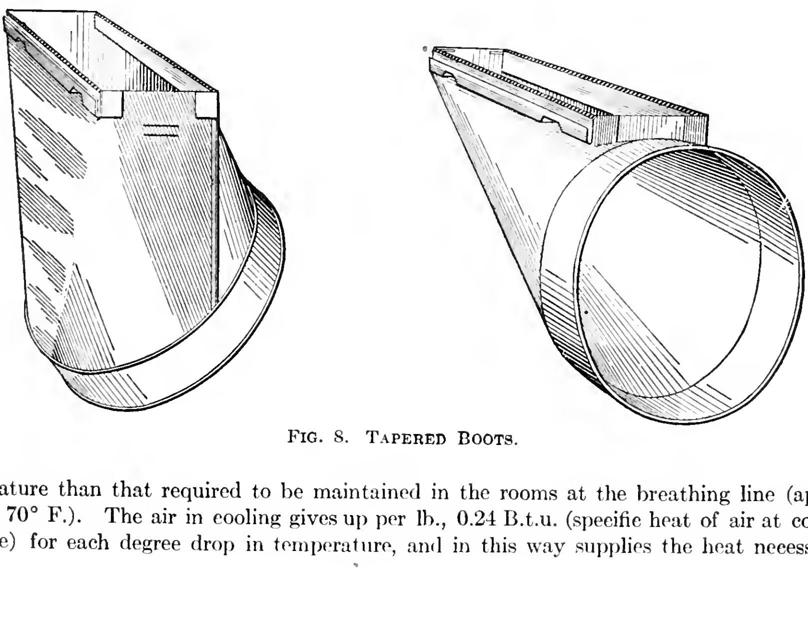


FIG. 8. TAPERED BOOTS.

temperature than that required to be maintained in the rooms at the breathing line (approximately 70° F.). The air in cooling gives up per lb., 0.24 B.t.u. (specific heat of air at constant pressure) for each degree drop in temperature, and in this way supplies the heat necessary to

offset the heat transmission of the walls, etc., and at the same time provides a supply of fresh air for ventilation. The maximum temperature of the air leaving the heater cap is approximately 180° F., and it leaves the registers at 150° F., allowing a drop in temperature of 30° between the furnace bonnet and the registers. These figures are maximum values not to be exceeded.

If the air is all drawn from the outside, and the outside temperature is 0°, then the air

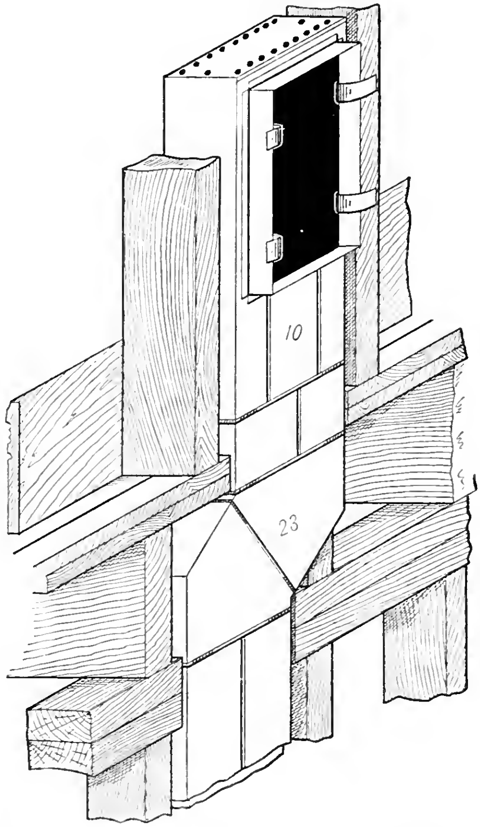


Fig. 9. Offset and Register Head Double Wall Pipe.

is heated from 0° to 180°, or 180°, and cooled, in entering the room, from 150° to 70° or 80°. In other words, 0.24 (180–80) or 24 B.t.u. is apparently thrown away, for every pound of air circulated.

If all the air must be brought in from the outside in order to supply a sufficient amount for ventilation, then this is the price which must be paid for ventilation, and it would be the same, no matter what system of heating is employed, for equally good ventilation. It is almost invariably the case, however, that a considerable portion of the air may, if desired, be recirculated, in which event, for equal ventilation effect, the furnace system of heating requires no more expenditure of heat in the form of fuel burned than a direct steam or hot-water system and is therefore just as economical to operate when correctly designed, installed and operated.

Head Producing Flow. The head producing the flow or circulation is due to the difference in weight between the ascending column of heated air and the weight of an imaginary column of the colder outside air. See detailed discussion in chapter on "Gravity-Indirect Heating."

The head or pressure available for overcoming the friction of the air ducts and creating the velocity of flow is comparatively feeble and consequently this system finds its greatest application in heating buildings of moderate size (1200 sq. ft. in ground area or less) in which the lengths of the air ducts are comparatively short.

In large buildings a fan is sometimes employed to give a positive circulation of the air. The use of fans in this connection is discussed in the chapter on "Hot Blast Heating."

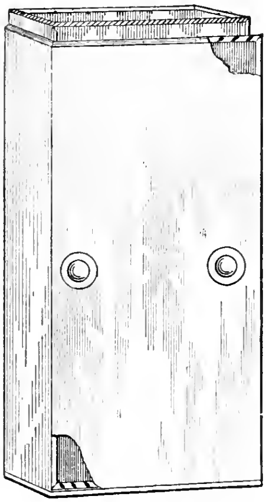


FIG. 10. SECTION OF DOUBLE WALL PIPE—2 FEET LONG.

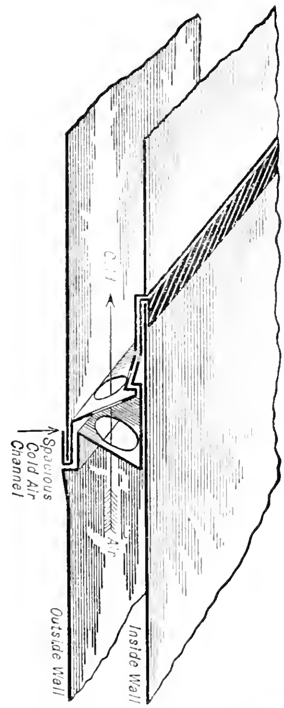


FIG. 11. LOCKING JOINT FOR DOUBLE WALL PIPE. (*Excelsior Steel Furnace Co.*)

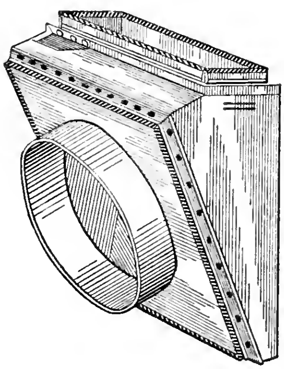


FIG. 12. BOOT OR FOOT PIECE FOR DOUBLE WALL PIPE.

Ventilating flues should be provided in the more important rooms for the escape of the air from the room; otherwise the circulation may prove unsatisfactory.

For a detailed discussion of the relation between the theoretical and actual velocities which may be expected in ventilating flues, see the chapter on "Gravity-Indirect Heating by Steam and Hot Water."

Pounds of Air to be Circulated per Hour. It is first necessary to determine the weight of air required per hour which must be supplied to each room.

Let W = lb. of air to be circulated per hour.
t = inside temperature to be maintained.
t_d = temperature of air leaving the registers (assumed 30° lower than the temperature leaving the furnace cap or bonnet).
H = B.t.u. to be supplied to room per hour as determined by heat loss calculations.
$0.24\ (t_d - t)$ = B.t.u. given up per lb. of air circulated.

$$\text{Then } W = \frac{H}{0.24\ (t_d - t)} \qquad (1)$$

The maximum value for t_d is 150° F. and t = 70° F., then

$$W = \frac{H}{19.2} \qquad (2)$$

d_o = density of outside air, at temperature t_o.
d_h = density of air leaving heater, at temperature t_h.
Q_1 = cu. ft. of outside air to be introduced per hour.
Q_2 = cu. ft. of warm air entering room per hour.

If outside temperature, $t_o = 0°$, then $d_o = 0.0863$, and if $t_h = 180°$, then $d_h = 0.0621$, and $t_d = 150°$, then $d_d = 0.0651$. Then

$$Q_1 = \frac{W}{d_o} = \frac{H}{1.64} \qquad (3)$$

$$Q_2 = \frac{W}{d_d} = \frac{H}{1.24} \qquad (4)$$

Heat Required from Heater per Hour. The heat required per hour from the heater will depend on the temperature of the entering air and will be a maximum when all the air circulated is taken from the outside.

Let h = B.t.u. required from heater per hour.
t_e = temperature, air entering heater = t_o = 0° F.
t_h = temperature, air leaving heater = 180°.

$$\text{Then } h = 0.24\ W(t_h - t_e) \qquad (5)$$

substituting the values given above for W, t_h, and t_e;

$$h = 2.25\ H \qquad (6)$$

Effect of Ventilation Requirements on Temperature at Register, and Weight of Air. Ordinarily the air, when all drawn from the outside, and introduced for heating any building, in which a warm-air furnace with natural circulation would be used, will cover the requirements for good ventilation. If, however, the installation must also meet certain ventilation requirements, then the amount as calculated from a consideration of the heat loss only, Equation (1), may not be sufficient and more air must be introduced at a lower temperature to prevent overheating.

If Q = cu. ft. required per hour for ventilation at a temperature of 70° = 1800 × number of persons,

d_t = density of air at 70° = 0.075 lb.

W_2 = lb. of air to be supplied per hour for ventilation = $Q\,d_t$.

If W_2 is found to be greater than W as calculated by Equation (1), then the following formula should be used.

$$W_2 = Q\,d_t = \frac{H}{0.24\,(t_d - t)} \qquad (7)$$

The final temperature of the air entering the room will be

$$t_d = \frac{H + 0.24\,Q\,d_t\,t}{0.24\,Q\,d_t} \qquad (8)$$

The temperature of the air leaving the heater t_h may be assumed 30° higher and substituting this value for $t_h = t_d + 30$ in Equation (5) we may obtain the heat to be supplied by the heater per hour, as already indicated.

Recirculation. If W_2 is found to be less than W, then some of the air may be recirculated with a consequent reduction in the heat expenditure and fuel consumption.

If W_2 = lb. of fresh air to be supplied per hour for ventilation

W = lb. of air to be circulated per hour from Equation (1)

$X = W_2/W$ = the proportion of air circulated that must be fresh air.

t = inside temperature.

t_1 = outside temperature.

t_e = temperature of the air entering the heater.

t_h = temperature of the air leaving the heater.

$$\text{Then } t_e = Xt_1 + (1 - X)t = X(t_1 - t) + t \qquad (9)$$

If $t_1 = 0°$ and $t = 70°$

$$\text{Then } t_e = 70(1 - X) \qquad (10)$$

The heat to be supplied by the heater will then be per hour

$$h = 0.24\,W(t_h - t_e) \text{ B.t.u.} \qquad (11)$$

From Equation (11) it is apparent that a smaller furnace could be used when recirculation is practiced. It is recommended, however, that the size of furnace be based on all outside air supply and the equivalent heat required as shown by Equation (6).

Size of Furnace. The capacity of a furnace for heating air depends primarily upon the amount of coal that may be burned per hour, which is the product of the *rate of combustion* times the grate area. With an assumed or fixed rate of combustion, the capacity of the furnace is dependent upon the grate area. The grate area is therefore used as a basis for the rating and comparison of warm air furnaces.

The average rate of combustion that is usual in furnace heating ranges from 3 to 5 lb. per sq. ft. of grate surface per hour, but in zero weather this rate may run as high as 8 lb., and is readily obtainable with the ordinary height of residence chimney; that is, at least 35 feet.

A properly designed furnace will have a combined furnace and grate efficiency of 60 to 70%, higher efficiencies having been obtained in tests.

Commercial Ratings of Furnaces. Manufacturers rate their furnaces according to the amount of space (cubical contents) in the ordinary residence construction they will heat to 70° F. in zero weather. (Maximum temperature of air leaving registers = 150° F.)

The detailed dimension and capacity data (other than grate area and space heated) of

most furnaces are seldom published by the manufacturer, although there are a few notable exceptions, as the information contained in the following table will show.

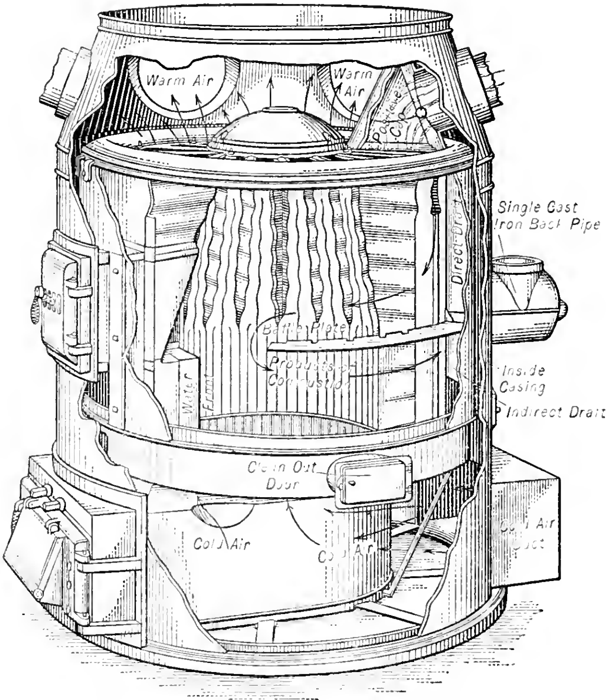


FIG. 13. KELSEY WARM AIR GENERATOR.

TABLE 2

DIMENSIONS AND CAPACITIES—KELSEY WARM AIR GENERATOR

(*Kelsey Heating Co.*)

See Fig. 13

Grates			Heating Surfaces			
Size or Number of Generator	Diam. of Grate and Fire Cylinder, Inches	Grate Area, Square Inches	No. Long Corrugated Heat Tubes	Heating Surfaces, Square Feet	Square Feet Heating Surfaces to Each Square Foot Grate Surface	Size of Smoke Stack, Inches
14	14	154	8	91	85	7
16	16	201	8	114	82	7
18	18	254	10	135	78	7
21	21	346	12	146	61	9
24	24	452	14	161	51	9
27	27	572.5	14	176	44	9
30	30	707	16	211	43	9

TABLE 2. (*Continued.*)

Cold Air Supply, Etc.				Dimensions		
Size or Number of Generator	Inside Diameter Brick Work if Pit Is Used	Size of Cold-Air Duct in Ins. if Air is Taken from Outside	Size of Cold-Air Face in Ins. if Air is Taken from Inside	Diam. of Base. Inches	Height of Castings. Inches	Height Generator Cased Complete. Inches
14	34	12 x 24	18 x 24	38	49	61
16	38	12 x 30	20 x 26	42	53	63
18	42	12 x 36	21 x 29	46	56	68
21	49	14 x 40	24 x 32	53	59	69
24	52	14 x 48	24 x 32	56	59	69
27	56	14 x 56 or (2) 14 x 32	30 x 36 or (2) 21 x 29	60	62	72
30	60	14 x 60 to 72 or (2) 14 x 40	30 x 48 or (2) 24 x 32	64	66	76

Size or Number of Generator	Measure'ts of Galv. Casings and Tops Series A, B, C, Inches	Weights of Generators	
		"Castings" Lb.	"Complete" Lb.
14	35	946	1008
16	40	1105	1168
18	43	1546	1635
21	$50\frac{1}{3}$	1924	2033
24	52	2189	2300
27	$56\frac{1}{2}$	2475	2600
30	$59\frac{1}{4}$	2994	3124

Size or Number of Generator	Heating Capacities					Size of Coal Recommended
	House Heating			Church Heating		
	Number of Average Size Pipes or Rooms	Estimated Capacity in Cubic Feet	Total Area of Heating Pipe Supplied by each Generator in Sq. Inches	Number of Pipes	Estimated Capacity in Cubic Feet	
14	3 to 4	4,000– 6,000	280 to 350	1	8000	Chestnut
16	4 to 6	6,000–10,000	350 to 420	1 to 2	10,000–14,000	Chestnut or Stove
18	6 to 8	12,000–16,000	450 to 500	1 to 2	16,000–20,000	Stove
21	9 to 11	18,000–24,000	575 to 625	1 to 2	25,000–35,000	Stove or Egg
24	10 to 13	24,000–32,000	675 to 750	1 to 3	35,000–45,000	Stove or Egg
27	12 to 15	35,000–42,000	850 to 925	1 to 3	50,000–60,000	Egg
30	14 to 19	45,000–55,000	975 to 1100	2 to 4	70,000–90,000	Egg

The actual size of the furnace naturally depends upon the heat transmission of the walls, floors, and roofs, plus the infiltration losses, as already explained. The claim, however, is made that these "in turn bear a reasonably uniform relation to the cubical contents of the ordinary house," with the usual proportions and ratios of wall to glass surface, and that therefore the rating, as given; is justifiable.

The two tables following were taken from the "Warm Air Furnace Handbook," published by the *Federal Furnace League*, an association of United States furnace manufacturers.

This "League" is no longer in existence. It has been succeeded by a stronger organization of progressive manufacturers, known as "*The National Warm Air Heating and Ventilating Association.*"

If the majority of the basement or leader pipes exceed 12 feet in length or have less than 1 inch

rise to the foot, or if the building has over one sixth its outside surface glass, then the furnace should be increased one or more sizes.

The size of the furnace required can also be determined by the combined area of the cross sections of the warm-air pipes. The heat pipes (leaders and stacks) are figured for an average approximate velocity of 6 ft. per second in the tables below.

These tables are based on a combustion rate of 8 lb. per sq. ft. of grate surface per hour, with 70° inside building, and + 10°, 0° and − 10° F. weather temperature outside. Temperature of air in furnace cap = 180° F., and air leaving registers = 150° F. All of the air is brought in from the outside to warm the building, and no recirculation allowed.

TABLE 3

CAPACITY OF WARM-AIR FURNACES OF ORDINARY CONSTRUCTION IN CUBIC FEET OF SPACE HEATED

(Federal Furnace League)

Divided Space—Cubic Feet			Fire Pot		Undivided Space—Cubic Feet		
+ 10°	0°	—10°	Diam. In.	Area Sq. Ft.	+ 10°	0°	—10°
12000	10000	8000	18	1.8	17000	14000	12000
14000	12000	10000	20	2.2	22000	17000	14000
17000	14000	12000	22	2.6	26000	22000	17000
22000	18000	14000	24	3.1	30000	26000	22000
26000	22000	18000	26	3.7	35000	30000	26000
30000	26000	22000	28	4.3	40000	35000	30000
35000	30000	26000	30	4.9	50000	40000	35000

TABLE 4

AIR HEATING CAPACITY OF WARM-AIR FURNACES

(Federal Furnace League)

Fire-Pot		Casing*	Total Cross Sectional Area of Heat Pipes, "a" Square Inch	No. and Size of Heat Pipes that May Be Supplied
Diam. Ins.	Area Sq. Ft.	Diam. Ins.		
18	1.8	30–32	180†	3-9″ or 4-8″
20	2.2	34–36	280	2-10″ and 2-9″ or 2-9″ and 2-8″
22	2.6	36–40	360	3-10″ and 2-9″ or 4-9″ and 2-8″
24	3.1	40–44	470	3-10″ and 1-9″ and 2-8″ or 2-10″ and 5-8″
26	3.7	44–50	565	5-10″, 3-9″ or 3-10″, 4-9″ and 2-8″
28	4.3	48–56	650	2-12″, 3-10″ and 3-9″ or 5-10″, 3-9″ and 2-8″
30	4.9	52–60	730	3-12″, 3-10″ and 3-8″ or 5-10″, 5-9″ and 1-8″

* The casing diameter should be such that the minimum cross sectional area M, between casing and radiator, will be at least 20% greater than the total cross sectional area of all the heat pipes, a, or $M = 1.2 \times a$ square inches.

† The B.t.u. that the furnace will supply, h, assuming all the air is taken from the outside at 0°, and raised to 180°, and used from 150° to 70° may be approximated by using a velocity through the heat pipe area of 6 feet per second. [Density at average temperature (180 + 150) / 2 or 165° = 0.0635.]

$$h = \frac{3600 \times 6 \times A}{0.0635} \times (180 - 0) \times 0.24 = 60{,}000\, A \text{ or } 416\, a \text{ B.t.u. per hour.}$$

A = Area heat pipes in sq. ft., and a = area in sq. in.

Furnace Rating Based on Efficiency and Rate of Combustion. The B.t.u. per hour that a furnace is capable of imparting to the air (not the room) may also be estimated from the grate area by assuming that the average coal used will contain approximately 12,000 B.t.u. per pound. A combined furnace and grate efficiency of 65%, and a maximum combustion rate of 8 lb. per sq. ft. of grate per hour for maximum conditions (coldest weather) are also usually assumed.

* Note.—The ratio of heating to grate surface depends on the heat transmission efficiency of the heating surface, which is usually taken as from 2,000 to 2,400 B.t.u. per square foot per hour. In this connection see a paper on "The Heat Analysis of a Hot Air Furnace" by *Prof. J. R. Allen* in the *Journal*, Jan., 1916, *A. S. H. and V. Engrs.*

Grate Surface Required. The area of the grate is readily calculated as soon as the heat to be supplied to the building per hour has been determined.

h = B.t.u. required from furnace per hour for heating the air.
H = B.t.u. to be supplied building per hour.
C = heating value B.t.u of coal per lb.
E = combined furnace and grate efficiency.
R = rate of combustion, lb. of coal per sq. ft. of grate surface per hr.
G = grate area in sq. ft.
$h = G \times C \times E \times R$.
Then h = area of grate in sq. ft. $\times$ 12,000 $\times$ 0.65 $\times$ 8. . . . . (12)
= 62,400 $\times G$ which is B.t.u. transmitted to the air passing the furnace.

$$\text{or} \quad H = \frac{62{,}400 \times G}{2.25} \qquad (12a)$$

from equation (6), which is B.t.u. available for heating.

The grate surface necessary to supply a given number of B.t.u. either at the furnace or at the register is readily found as follows:

$$G = \frac{h}{C \times R \times E} \qquad (13)$$

$$\text{or} \quad G = \frac{2.25 \times H}{62{,}400} \qquad (13a),$$

since $h = 2.25H$.

Usual assumptions are:
C = 12,000 B.t.u. per lb.
R = 4 to 5 lb. ordinary rate and 8 lb. for maximum conditions in coldest weather.
E = 0.60 to 0.70.

$$\text{Then} \quad G = \frac{h}{12{,}000 \times 8 \times 0.65} = \frac{h}{62{,}400} \qquad (14)$$

The ratio of heating to grate surface ranges from 12 to 1, to 30 to 1 in most of the commercial furnaces now on the market.

Size of Leaders and Stacks. The area of the air pipes (leaders and stacks) required for a room depends upon the quantity of air to be introduced per minute and the velocity with which the air will flow with natural circulation.

$Q_2/60$ = cu. ft. warm air to be introduced into the room per minute.
V = velocity of air ft. per minute attainable.
A = area pipe sq. ft.
$Q_2/60 = AV$, and substituting value of $Q_2 = H/1.24$,
$A = H/74.4 \times V$ sq. ft. . . . . (15)

It is assumed that the following velocities are approximately obtained in the leaders and stacks for the floors as stated:

1st floor—4.5 ft. per sec. or 270 ft. per min.
2nd floor—6.5 ft. per sec. or 390 ft. per min.
3rd floor—8.0 ft. per sec. or 480 ft. per min.

The above velocities have been observed in practice in well designed systems.

Then for various floors substituting in Equation (15):

$$A_1 = H/20{,}088 \text{ 1st floor pipes, leaders and stacks.}$$
$$A_2 = H/29{,}016 \text{ 2nd floor pipes, leaders and stacks.}$$
$$A_3 = H/35{,}712 \text{ 3rd floor pipes, leaders and stacks.}$$

Actual leader and stack sizes are based on the above areas, using nearest half inch for leader diameter (Table 5), and keeping the stacks of such proportions that the cross sectional dimensions are never in a greater ratio than 3 to 1.

For example a stack 4″ × 20″ is seldom effective over its full area, being too narrow, and due to its large rubbing surface causing excessive friction.

The actual velocities obtained, however, will depend upon the head or pressure causing the flow and the friction head and will seldom exceed 50% of the theoretical. See chapter on " Gravity-Indirect Heating " for determination of head available and friction.

The following table has been recommended by the *Federal Furnace League* for various sizes of rooms. This table gives the sizes of round pipe for leaders, the size of wall pipe for stacks, and free areas of registers to connect with same. Leaders over 12 feet in length should be increased 1 in. in diameter for each 5 feet beyond 12 feet:

TABLE 5

CAPACITIES AND DIMENSIONS OF WARM AIR PIPING AND REGISTERS

(*Federal Furnace League*)

1	2	3	4
Diameter of Round Cellar or Riser Pipe, Inches*	Proper Size of Rectangular Riser Pipe Inches.†	Area of Riser Pipe Square Inches	Required "Free Area" of Register Face, Square Inches‡
6	3 x 9½	28	52
6½	3½ x 9½	33	62
7	3½ x 11	38	72
7½	3½ x 12½	44	84
8	3½ x 14	50	96
8½	4 x 14	57	108
9	4 x 16	64	120
9½	4 x 18	71	134
10	4 x 20	78	142
10½	6 x 14½	86	158
11	6 x 16	95	176
11½	6 x 17½	104	194
12	6 x 19	113	204
12½	6 x 20½	122	222
13	6 x 22	132	242
13½	8 x 18	143	254
14	8 x 19	154	276
14½	8 x 20½	165	298
15	8 x 22	176	320
16	8 x 25	201	358
17	10 x 22½	227	410
18	10 x 25½	254	450
19	12 x 23½	283	508
20	12 x 26	314	554
21	12 x 28½	346	618
22	14 x 27	380	686
23	14 x 29½	415	707
24	14 x 32	452	770

* When the required size of pipe falls on the odd half-inch (as 7½, 8½, 9½, etc.,) the size may be increased to the even inch (as 8 instead of 7½, 9 instead of 8½, etc.), for the first-floor rooms and bathrooms; provided that the pipes for upper-floor rooms, other than bathrooms, be decreased by one half inch when the required sizes fall on the odd half-inch. It is better, however, to use pipes of the sizes given in the above table, with proper allowances for length of pipe, extra bends, etc., beyond straight runs 12′-0″ long.

† See Table 6 for dimensions of Double Wall Pipe. Stacks with dimension ratios greater than 3:1 are to be avoided if possible, or next size larger used.

‡ Column 4 applies to registers with 55% "free area" and the area as listed provides for same velocity through "free area" of register grille as in stack.

Sizes and capacities of double and single wall pipe as manufactured are given in the following tables:

TABLE 6

DIMENSIONS OF EXCELSIOR DOUBLE WALL PIPE

(Adopted Dec. 1, 1912—Excelsior Steel Furnace Co.)

Number	Measurements, Inches			Area Stack Sq. In.	Collar Diameter Inches	Area Collar Square Inch	Register Size Square Inch Convex or Wafer
	Nominal	Inside	Outside				
1	3 x 10	$2\frac{3}{8}$ x 10	3 x $10\frac{5}{8}$	24	7	39	6 x 8— 8 x 10
6	3 x 12	$2\frac{3}{8}$ x 12	3 x $12\frac{5}{8}$	$28\frac{1}{2}$	8 and 9	51 and 63	8 x 10— 9 x 12
7	4 x 11	3 x 10	$3\frac{5}{8}$ x $10\frac{5}{8}$	30	8 and 9	51 and 63	8 x 10— 9 x 12
8	4 x 13	3 x 12	$3\frac{5}{8}$ x $12\frac{5}{8}$	36	8, 9 and 10	51 and $78\frac{1}{2}$	8 x 10—10 x 14
9	4 x 14	3 x 13	$3\frac{5}{8}$ x $13\frac{5}{8}$	39	9 and 10	63 and $78\frac{1}{2}$	10 x 12—10 x 14
12	6 x 13	5 x 12	$5\frac{5}{8}$ x $12\frac{5}{8}$	60	9 and 10	63 and $78\frac{1}{2}$	10 x 12—10 x 14
14	$6\frac{1}{2}$ x 14	$5\frac{1}{2}$ x 13	$6\frac{1}{8}$ x $13\frac{5}{8}$	72	10 and 12	$78\frac{1}{2}$	10 x 14—12 x 14

TABLE 7

DIMENSIONS OF EXCELSIOR SINGLE FURNACE PIPE

(Excelsior Steel Furnace Co.)

Measurement, Inches	Size of Boot Collars Diam., In.	Capacity of Collars Square In.	Capacity of Pipe Sq. In.
3 x 10	8	51	30
$3\frac{1}{2}$ x 10	8	51	35
3 x 12	8 and 9	51 and 63	36
$3\frac{1}{2}$ x 12	9	63	42
$3\frac{1}{2}$ x 13	9 and 10	63 and 78	45
$5\frac{1}{2}$ x 12	10	78	66
$5\frac{1}{2}$ x 14	12	114	77
$5\frac{1}{2}$ x 16	12 and 14	114 and 154	88
$7\frac{1}{2}$ x 16	14	154	112

NOTE.—Stacks $5\frac{1}{2}''$ deep made to order only. With frictionless boots, collars in same can be made of diameter equal to width of stack. Collars 11″ in diameter furnished when so ordered.

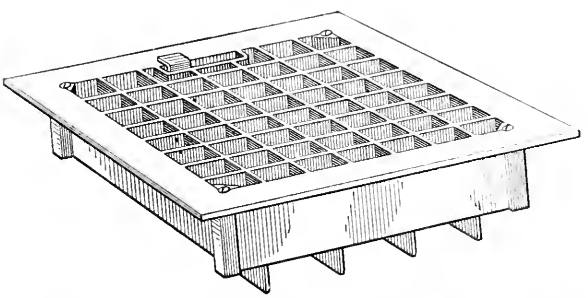


FIG. 14. FLOOR REGISTER WITH LOUVERS MADE OF SEMI-STEEL. *(Auer Register Co.)*

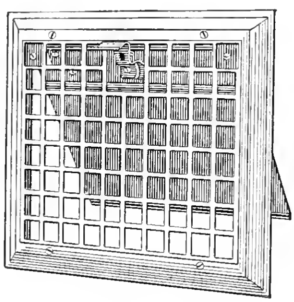


FIG. 15. CONVEX SIDE-WALL REGISTER (TO SET HORIZONTALLY).

Registers. The *free area* through the ordinary register grille is only approximately 50% of the gross area, consequently a register must be selected that has a gross area of double

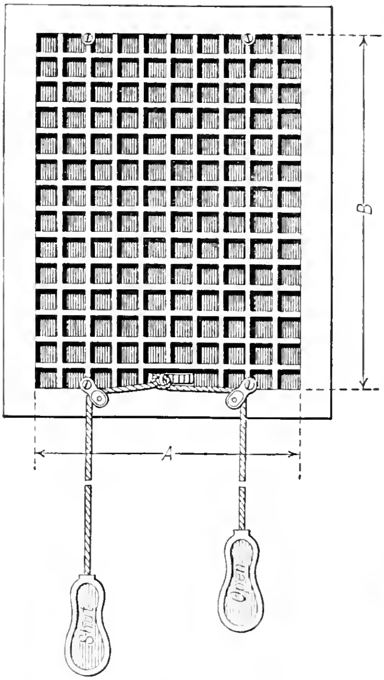


FIG. 16. SIDE-WALL REGISTER WITH RECTANGULAR GRILLE AND CONTROL CORDS.

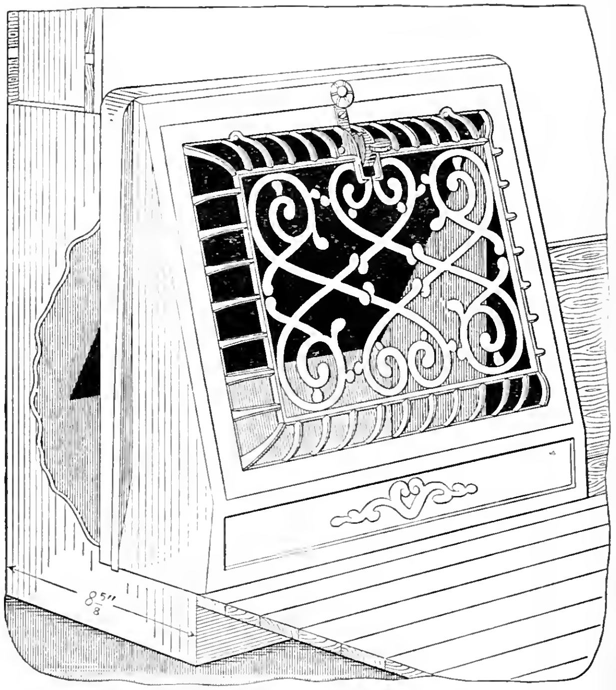


FIG. 17. BASEBOARD REGISTER FOR FIRST FLOOR WITH $4\frac{1}{2}$-INCH FLANGE.

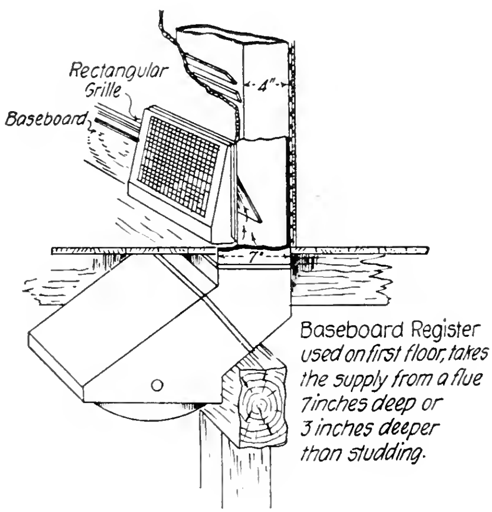


FIG. 18.

the area of the pipe with which it connects, in order that the air passage may not be contracted and the capacity reduced. Commercial register sizes are based on the actual inside dimensions of the grilled opening as "A" and "B" in Fig. 16, and are made either of pressed steel or cast iron, with a variety of fancy or plain grilles. The plain rectangular grille is to be preferred, finished to suit the decorative scheme, in black japan or electro-plated in brass, bronze or copper finish.

Warm air registers may be placed in the floor (Fig. 14), but preferably in inside partitions, for first floor rooms. By using the modern *baseboard register* (Figs. 17 and 18) it is usually possible to secure the required capacity without resorting to floor registers. These baseboard registers can be connected to a flue from 3 to 4½ inches deeper than the studding. This has been accomplished by making the special baseboard register so that it projects 2 inches into the room at the floor line, necessitating the floor being cut out, and also utilizing the space of about 1-inch occupied by the lath and plaster, or a total increase in depth of flue of about 3 inches. For upper floor rooms registers should be placed in inside partition walls, using *convex registers* (Fig. 15) for shallow stacks. As a general rule warm air registers should be so placed as to shorten leader and stack connections as much as possible.

The use of a floor register may be permitted in an entrance hall for drying shoes and garments, but it is most unsanitary and cannot fail to collect dirt and filth of all kinds, and is subject to the further objection that its use frequently requires the cutting of carpets. In case such registers are used, however, suitable *register boxes* (Fig. 19) must be provided, and are preferably constructed with double walls (Figs. 20 and 21).

TABLE 8

TABLE OF SIZES OF FLOOR REGISTERS, BASEBOARD REGISTERS AND REGISTER BOXES

Size of Round Cellar Pipe, Inches	Size of Round Floor Register, Inches	Size of Rectangular Floor Register,* Inches	Size of Register Box to Baseboard Where Studs Are Not More than 4 inches Deep Inches	Size of Baseboard Register Where Studs Are Not More Than 4 Inches Deep, Inches	Size of Register Box to Baseboard Register Where There Is No Limit to Depth of Register Box, Inches	Size of Baseboard Register Where There Is No Limit to Depth of Register Box, Inches
6	9	8 x 8	2¾ x 10	7 x 10	2¾ x 10	7 x 10
6½	9	8 x 8	3¼ x 10	7 x 10	3¼ x 10	7 x 10
7	10	8 x 10	3¾ x 10	7 x 10	3¾ x 10	7 x 10
7½	12	8 x 12	4¼ x 10	7 x 10	4¼ x 10	7 x 10
8	12	8 x 12	4¼ x 12	7 x 12	4¼ x 12	7 x 12
8½	12	9 x 12	4¾ x 12	7 x 12	4¾ x 12	7 x 12
9	14	10 x 12	5 x 13	8 x 13	5 x 13	8 x 13
9½	14	10 x 14	6 x 12	10 x 12	6 x 12	10 x 12
10	14	10 x 16	6½ x 12	10 x 12	6½ x 12	10 x 12
10½	16	10 x 16	6½ x 13	10 x 13	6½ x 13	10 x 13
11	16	12 x 15	6¾ x 14	12 x 14	6¾ x 14	12 x 14
11½	16	12 x 18	7 x 15	12 x 15	7½ x 14	12 x 14
12	18	12 x 20	6½ x 18	12 x 18	7½ x 15	12 x 15
12½	18	14 x 16	6¾ x 18	12 x 18	6¾ x 18	12 x 18
13	18	14 x 18	....	....	7¼ x 18	12 x 18
13½	18	14 x 20	....	....	8 x 18	12 x 18

*Note.—For additional register sizes see Table 9.

Ventilating Flues. The area of the ventilating flue is usually made approximately 0.80 that of the heat pipe or pipes supplying the room, as the volume of air to be handled is less owing to the lower temperature of the air leaving the room. Ventilating flues are often connected only with the downstairs and upstairs hallways, and fireplaces are always utilized as vents when they are installed. If ventilating flues are installed it is advisable to equip them with registers near the floor and ceiling of the room, the former for winter and the latter for summer use. All vent flues, including fireplaces, must have suitable dampers.

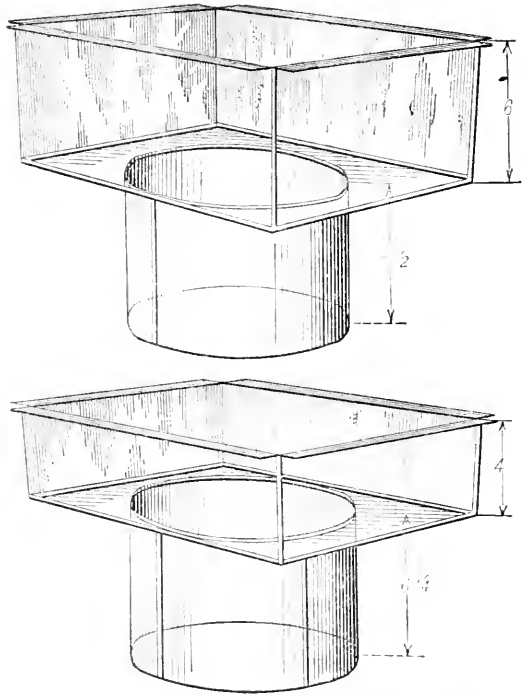


FIG. 19. EXCELSIOR SINGLE REGISTER BOXES.

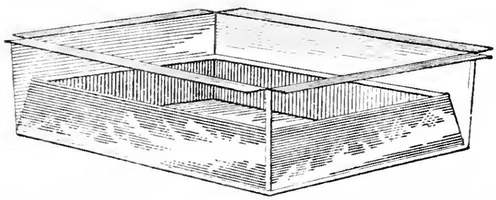


FIG. 20. EXCELSIOR DOUBLE SECOND FLOOR REGISTER BOX.

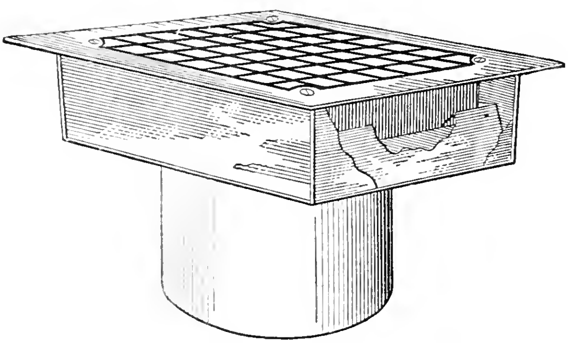


FIG. 21. EXCELSIOR DOUBLE FIRST FLOOR REGISTER BOX AND REGISTER.

The use of heat or vent registers near the ceiling will require that suitable chains or cords be connected to the valve or louver arm in order to control them conveniently. (Fig. 16.)

TABLE 9

STANDARD HEAT AND VENT REGISTER SIZES

(See Fig. 16)

Size of Opening A x B			
4 x 10	8 x 10	14 x 20	22 x 24
4 x 12	8 x 12	14 x 22	24 x 24
4 x 15	9 x 12	16 x 18	24 x 27
5 x 10	10 x 12	16 x 20	24 x 30
5 x 12	10 x 14	16 x 24	27 x 27
5 x 15	12 x 12	18 x 21	27 x 38
6 x 8	12 x 14	18 x 24	28 x 30
6 x 10	12 x 15	20 x 20	30 x 30
6 x 12	12 x 19	20 x 24	30 x 36
6 x 16	14 x 16	20 x 26	36 x 36
6 x 24	14 x 18	21 x 29	

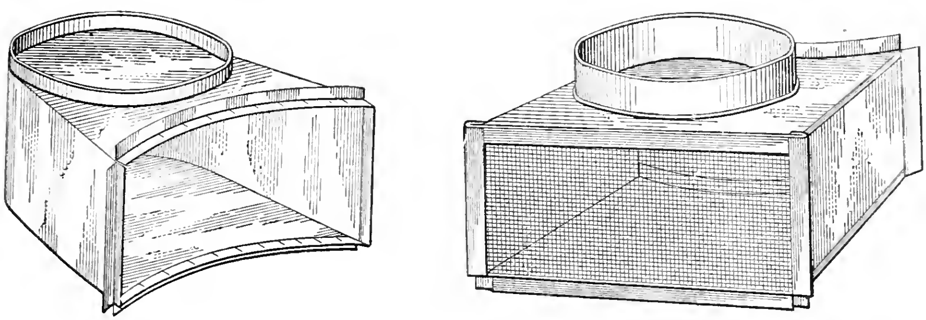


FIG. 22. COLD AIR SHOES.

Fresh Air Duct. The area of the cold air duct (Figs. 23 and 24) may be made 0.75 of the combined areas of the leader pipes, this being about the ratio of volumes at the respective temperatures. The fresh air duct may be run either above or below the floor, and if above it may be constructed of galvanized sheet steel, or of matched well seasoned *T* and *G* lumber and connection made to the furnace through a suitable *cold air shoe* (Fig. 22). All cleats should be placed on the outside of the box.

If below the floor, brick, terra-cotta pipe, or concrete construction may be used, and connection made to a water tight cold-air pit beneath the furnace.

It is most essential that all fresh air pipes as well as the furnace jacket be air tight to prevent the dust from the coal and ashes passing through the air ducts into the rooms.

It is always desirable to provide a *fresh air room* (Fig. 24) with suitable baffle plates to assist in the removal of dust or dirt. In no case should screens be used as filters on account of friction losses through the material of same. These fresh air rooms should have a net area of about six times the area of the fresh air duct to reduce the velocity of the incoming air sufficiently to allow the dust to settle out.

Recirculating Duct. The use of recirculating ducts, through which part or all of the air delivered to the rooms is returned to the furnace to be reheated, will greatly reduce the cost of operation.

The combined *area of the recirculating ducts* will depend upon the amount of air that may be recirculated as determined by calculation. Assume that the average velocity will be the same

as the average velocity through the leader pipes, and neglect the difference in densities between the hot air, and the air entering the recirculating duct. Then if $X =$ the fractional part of the whole amount of air circulated that is to be fresh air and A = combined area of leader pipes and A_R = the area of the recirculating ducts, we have $A_R = (1 - X)A$, or more exactly $A_R = \frac{(1 - X)W}{6 \times 3600 \times 0.075}$. Where W = pounds of air to be circulated per hour, 6 = average velocity in feet per second, assumed the same as the average velocity through all the leader pipes, and 0.075 = density at 70° the temperature of recirculated air.

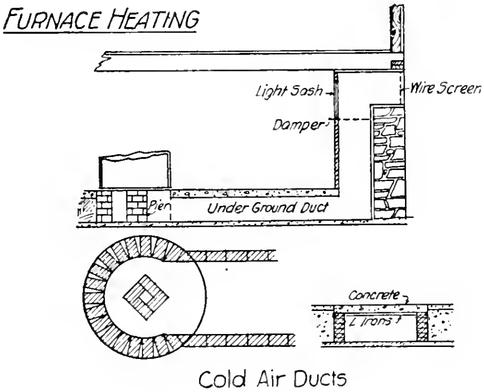


Fig. 23.

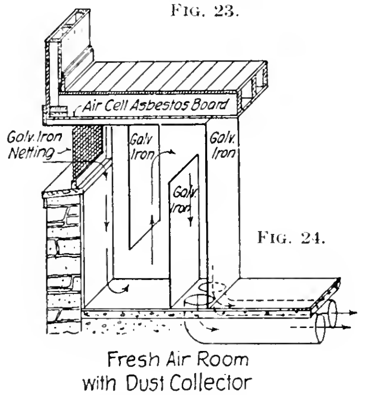


Fig. 24.

Figs. 23 and 24.

Register faces (Fig. 25) for recirculating ducts are generally made of hardwood with a free area about 50% of the gross or listed area (Table 11). These registers are often placed only in first floor hall, either in the floor or preferably at side of stairway or in the face of a hall seat (Fig. 26).

Since this recirculated air is already at 70°, less heat is required to raise one pound of it to 180° than would be needed if all outside air was used at 0° F.

These faces are made of oak strips ⅜ of an inch thick by 1¼ inches in width, into which cross strips ⅜ of an inch square are glued in grooves made to receive them. The square strips are let into the larger strips in such a manner as to make the top of the face even. The meshes are 1⅛ inches square.

TABLE 10

TABLE OF CAPACITIES AND DIMENSIONS OF FRESH AIR DUCTS, ROOMS, ETC.

Size of Horizontal Portion of Rectangular Fresh Air Duct, Inches	Size of Horizontal Portion of Round Fresh Air Duct, Inches	Cross Section Area of Horizontal Portion of Fresh Air Duct, Inches	Size of Fresh Air Room—Length and Width (Height Same as Depth of Cellar), Inches	Size of Fresh Air Intake (Area of Woven-wire Netting Not Including Frame), Inches
8 x 18	1–14	144	18 x 48	12 x 16
8 x 21	1–15	168	21 x 48	14 x 16
8 x 24	1–16	192	24 x 48	16 x 16
10 x 21	1–16	210	21 x 60	14 x 20
10 x 24	1–18	240	24 x 60	16 x 20
10 x 27	2–13	270	27 x 60	18 x 20
10 x 30	2–14	300	30 x 60	20 x 20
12 x 27	2–14	324	27 x 72	18 x 24
12 x 30	2–15	360	30 x 72	20 x 24
12 x 33	2–16	396	33 x 72	22 x 24
12 x 36	2–17	432	36 x 72	24 x 24
12 x 39	2–17	468	39 x 72	24 x 26
14 x 36	2–18	504	36 x 84	24 x 28
14 x 39	2–19	546	39 x 84	26 x 28
14 x 42	2–19	588	42 x 84	28 x 28
14 x 45	2–20	630	45 x 84	28 x 30
14 x 48	2–21	672	48 x 84	28 x 32
14 x 51	2–21	714	51 x 84	28 x 34
16 x 48	2–22	768	48 x 96	32 x 32
16 x 51	2–23	816	51 x 96	32 x 34
16 x 54	2–24	864	54 x 96	32 x 36
16 x 57	2–24	912	57 x 96	32 x 38
16 x 60	2–25	960	60 x 96	32 x 40

TABLE 11

SIZES AND CAPACITIES OF WOOD REGISTER FACES FOR COLD AIR DUCTS

Size	Net Area of Air Space in Square Inches	Nearest Size of Round Pipe of Equivalent Area	Size	Net Area of Air Space in Square Inches	Nearest Size of Round Pipe of Equivalent Area
		In.			Ins.
12 x 20	135	12	24 x 24	323	20
12 x 24	161	14	24 x 26	349	20
12 x 30	202	16	24 x 30	403	22
14 x 20	157	14	28 x 28	439	22
14 x 26	203	16	30 x 30	504	26
16 x 20	179	14	36 x 20	403	22
16 x 24	215	16	36 x 24	484	24
16 x 30	269	18	36 x 30	605	28
18 x 24	242	18	36 x 36	725	30
18 x 30	303	20	72 x 18	726	..
20 x 20	224	16	72 x 20	806	..
20 x 24	269	18	72 x 24	968	..
20 x 26	291	18	72 x 30	1210	..
20 x 30	336	20	72 x 36	1450	..

Fuel Consumption. The total fuel consumption for the heating season, usually considered from October 1st to May 1st, may be approximated as follows, when the system has been proportioned for 70° F. inside in zero weather.

h = B.t.u. to be supplied by furnace per hour in zero weather.

t_a = average outside temperature during the heating season for the locality (see table in chapter on "Heat Transmission of Building Materials," or *U. S. Weather Bureau* records).

t_1 = average inside temperature during the same period (approximately 65°).

C = calorific value of fuel per lb.

E = combined efficiency of furnace and grate (will not average over 45% for the season).

S = total number hours in heating season = 5,088 (Oct. 1st to May 1st).
T = tons of coal (2,000 lb.) per season.

$$T = \frac{h/70 \times (65 - t_a) \times S}{C \times E \times 2000}.$$

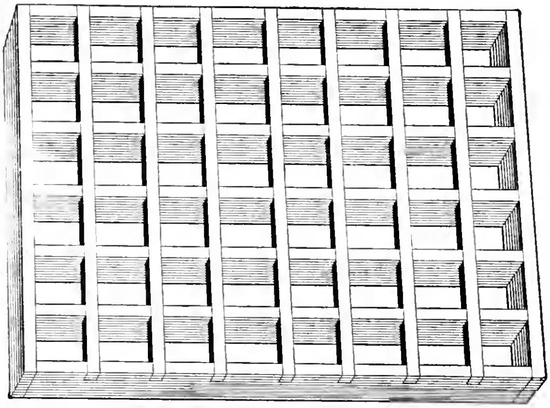


Fig. 25. Wooden Cold-Air Face.

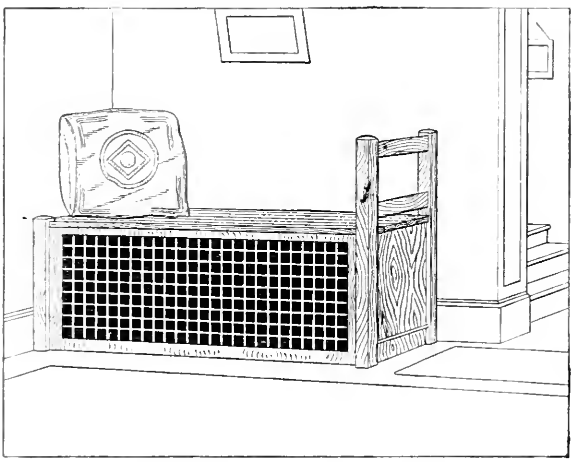
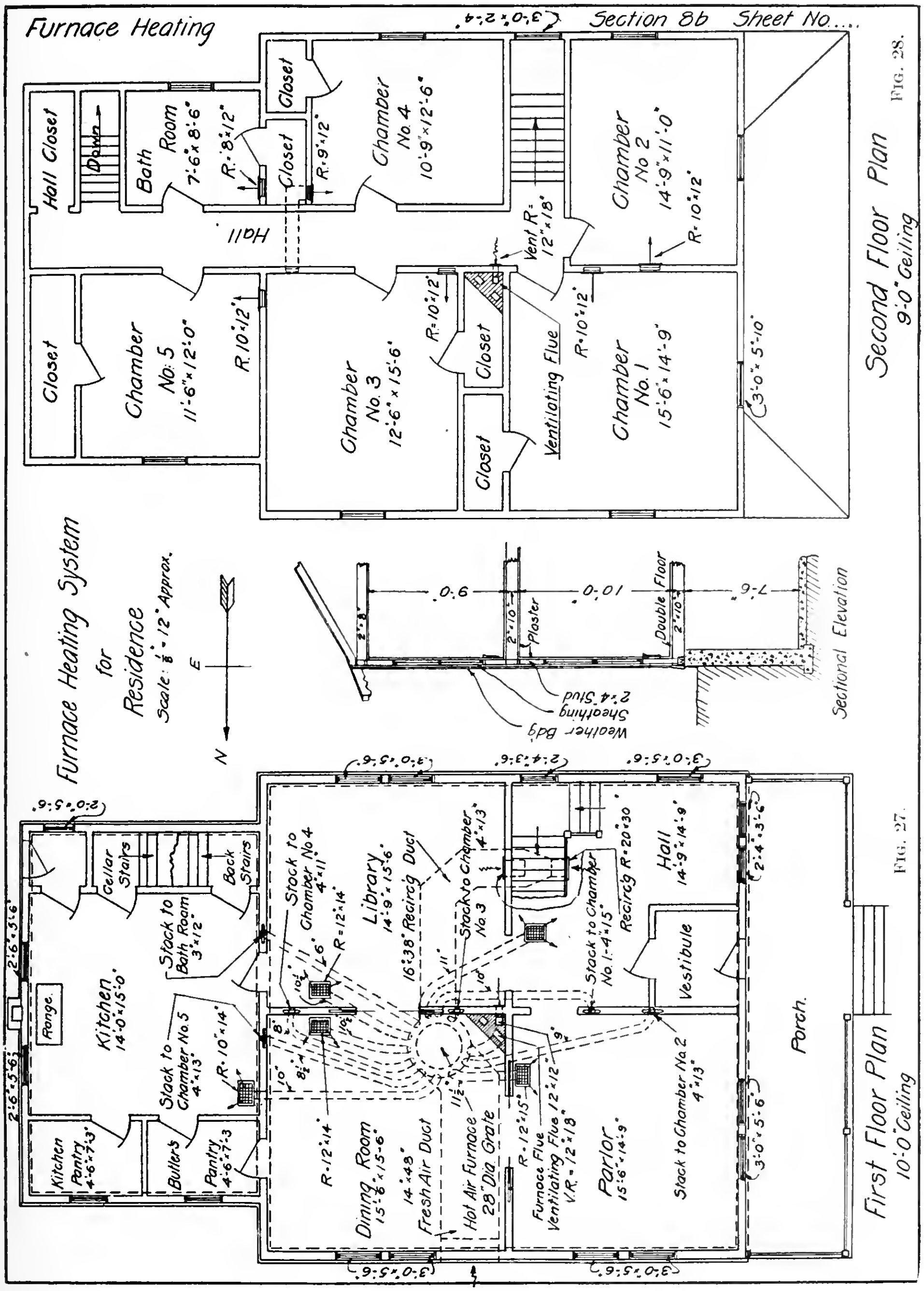


Fig. 26. Wooden Cold-Air Face in Hall Seat.

FURNACE HEATING PROBLEM

A gravity furnace heating system is to be designed for the two-story frame building shown on plans, (Figs. 27 and 28) with inside and outside temperatures of 70° and 0° F. respectively. No allowance is to be made for exposure. Transmission and infiltration losses are as computed in Table 12, which also gives size of heat pipes, leaders and stacks, and register sizes.

Size of Furnace and Grate. The size of furnace is calculated on the assumption that all the air is taken from the outside. The total calculated heat loss from the building per hour is 124,779



Figs. 27 and 28.

Furnace Heating - Data and Calculations

No allowance for exposure – Infiltration based on 1/32" crack or 1.2 B.t.u. per 1'-0" per 1° Fah

TABLE 12. (SEE FIGS. 27 AND 28)

First Floor

Surface or Description	B.t.u per hour 1 □	Parlor 2280 Cu. ft.		Hall 2170 cu. ft.		Dining Room 2400 cu. ft		Library 2280 cu. ft.		Kitchen 3600 cu. ft.	
		Sq. ft.	H	Sq. ft.	H.	Sq. ft.	H	Sq. ft.	H	Sq. ft.	H
1	1,a	2	3	4	5	6	7	8	9	10	11
Wall	22.4	253	5667	253	5667	157	3517	157	3517	471	10550
Glass	70.0	50	3500	41	2870	33	2310	33	2310	69	4830
Infiltration	1.2×70 =84	33	2838	25	2150	33	2838	33	2838	275	2365
Floor	12.5	220	2850	217	2713	240	3000	228	2850	360	4500
Total H	—		14855		13400		11665		11515	½ ×	22245
Area of Heat Pipe □"	H/20088	.743 □"		.670 □"		.584 □"		.575 □"		.557 □"	
Diam. Leader		11½"		11"		10½"		10½"		10"	
Reg. size of Face		12" × 15"		12" × 14"		12" × 14"		12" × 14"		10" × 14"	

Leader Pipes

Diam.		Areas
11½	=	103.8
11	=	95.0
10½×2	=	173.2
10×2	=	157.0
9	=	63.6
8½×2	=	113.4
8	=	50.3
6	=	28.3
Total	=	784.6
		.75
75%	=	589

h - 2.25 H = 270,753 B.t.u. H = Total heat loss - 124,779 B.t.u.

Second Floor

Surface or Description	B.t.u. per hour 1 □	Chamber #1 2052 cu. ft.		Chamber #2 1458 cu. ft.		Chamber #3 1746 cu. ft.		Chamber #4 1206 cu. ft.		Chamber #5 1242 cu. ft.		Bath Room 576 cu. ft.	
		Sq. ft.	H	Sq. ft.	H	Sq. ft.	H	Sq. ft.	H	Sq. ft.	H	Sq. ft.	H
1	1, a	2	3	4	5	6	7	8	9	10	11	12	13
Wall	22.4	237	5308	197	4413	149	3333	149	3333	226	5062	59	1322
Glass	70.	35	2450	35	2450	17.5	1225	17.5	1225	17.5	1225	17.5	1225
Infiltration	1.2×70 =84	17.5	1505	17.5	1505	17.5	1505	17.5	1505	17.5	1505	17.5	1505
Ceiling	22.4	228	5107	162	3631	194	4345	134	3002	138	3091	64	1434
Total H	—		14370		11999		10413		9070		10883		5486
Area of Heat Pipe □"	H/29016	.496		.414		.359		.312		.375		.189	
Diam. Leader		10"		9		8½"		8"		8½"		6"	
Stack (Dble. Wall Pipe)		4" × 15"		4" × 13"		4" × 13"		4" × 11"		4" × 13"		3" × 12"	
Reg. size of Face		10" × 14"		10" × 12"		10" × 12"		9" × 12"		10" × 12"		8" × 12"	

Total cu. ft (incl. 2nd fl. Hall) = 26,000 – Total area, sq. in of heat pipes = 785 □". Recirculating duct = 0.8 × 785 = 604 □" = 16" × 38" and 2-20" × 30" Registers 50% free area. – Grate area = 270,753 ÷ 62,400 = 4.34 □' or Diam. = 28" and cold air duct = 785 × 0.75 = 589 □" = 14" × 48" — Assumed kitchen range supplies ½ heat for Kitchen, Pantry, etc. = = 22,245 ÷ 2 = 11,123 B.t.u. per hour.

TABLE 12.

B.t.u., which, divided by 62,400, the heat available from 1 sq. ft. of grate when burning 8 lb. of coal, of 12,000 B.t.u. heat value per pound, at 65% efficiency, gives 4.34 sq. ft. as the grate area. This will require a grate of 28″ diameter.

This building has a net volume of 26,000 cu. ft. and by reference to Table 12 it will be seen that a 28″ grate is recommended for this amount of divided space.

The grate has been proportioned on the assumption that 8 lb. of coal can be burned per sq. ft. per hr. in zero weather, with an over-all efficiency of 65%.

The infiltration loss is based on a 13-mile wind velocity in zero weather, with leakage through a crack at openings $\frac{1}{32}$″ wide, for which the B.t.u. loss is 1.2 per lineal foot per hour per 1°.

The furnace in this problem has been located practically in the center of the house, but

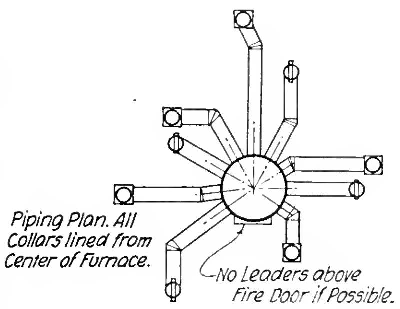


FIG. 29.

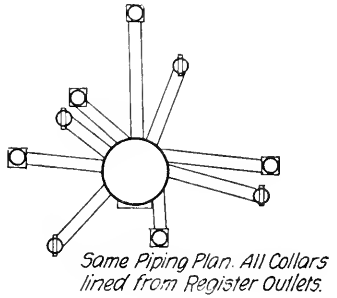


FIG. 30.

on the north side of its east and west axis, giving a direct cold air connection from the north wall and short direct runs for most of the leaders.

Leader Layout. The leaders may be laid off, as shown here and in Fig. 29, by dividing up the circumference of the bonnet into areas proportional to the amount of air to be distributed by each leader, and then connecting collar and leader radially to furnace cap, making one or more elbows in the leader, if necessary, to connect with stack. Another method is to run practically all leaders direct from furnace to foot of stack (Fig. 30) and cut the collars in on the angles at which they intersect the casing. The former method is recommended, and requires less skill in installation.

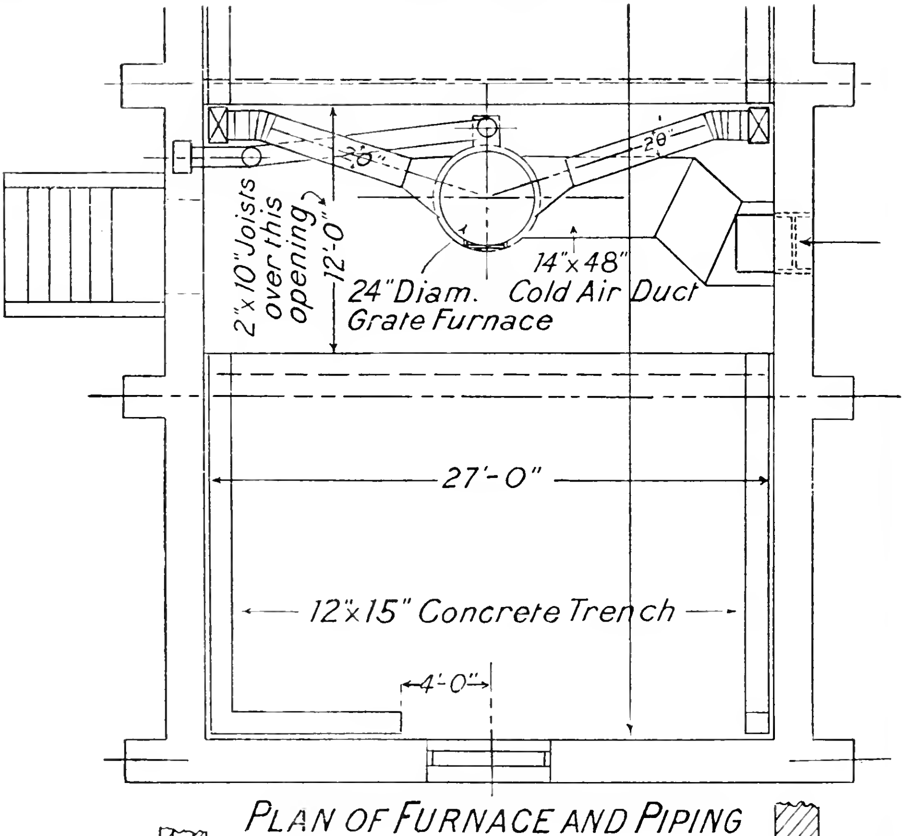
The basement heating plan (Fig. 27) is shown on the first floor plan, which also shows all stack sizes to both floors.

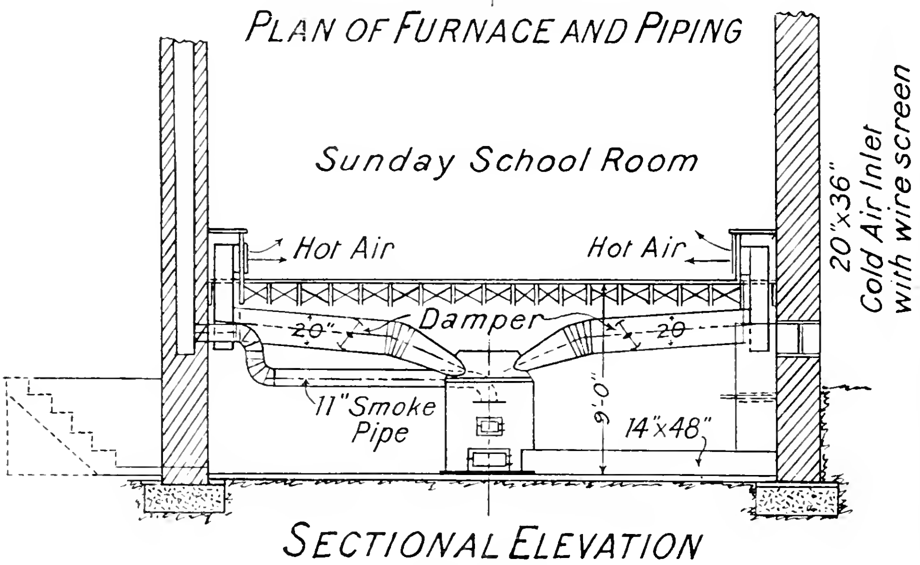
NOTE.—Floor registers have been shown on the first floor plan in order to simplify the layout and make the plan clearer. In general, baseboard registers are to be preferred.

Furnace Heating System for a Church. A typical, although somewhat unusual, application of a furnace heating system is shown in Figs. 31 to 33, in which the heated air is distributed through two very large leaders to the audience room of a small church. The method of connecting these leaders to the furnace bonnet is a departure from the usual practice, as well as the use of vertical registers placed under seats or boxes.

HUMIDITY IN FURNACE HEATING

Necessity for Humidification. The moisture carrying capacity of air increases rapidly with the temperature, and hence when the outdoor air is warmed to 70° in cold weather the amount of moisture originally present in the air is insufficient to maintain a *reasonable humidity* at the higher temperature. The term "reasonable" refers to a moisture content between 40 and 50% of complete saturation, such that there will be no danger of condensation on windows and other cold surfaces.





FIGS. 31 AND 32.

NOTE.—Provision has been made for heating this building by steam at a later date, and for this purpose the necessary return trenches have been provided under the floor which extends over the unexcavated portion of the building.

Example. (Case 1.) If air at 32° and 50% relative humidity is heated to 70° and no moisture added at the same time, we will have a humidity of 13.2%. As the maximum moisture carrying capacity of each cubic foot is 2.128 grains at 32°, and 8.071 grains at 70°, the relative humidity at 70° = $\frac{0.5 \times 2.128}{8.071} \times 100 = 13.2\%$.

Refer to table on "Properties of Air and Vapor Mixtures," in chapter on "Air Conditioning."

If it is desired to keep the inside humidity at 50% and we find there are 10,000 cu. ft. of air entering the room at a register temperature of 150° F., we must first find the volume of this air at 32° and 70°.

$$10{,}000 \times \frac{530}{610} = 8{,}700 \text{ cu. ft. at } 70°.$$

$$8{,}700 \times \frac{492}{530} = 8{,}100 \text{ cu. ft. at } 32°.$$

The values 492, 530, and 610 are the absolute temperatures corresponding to 32°, 70° and 150° F.

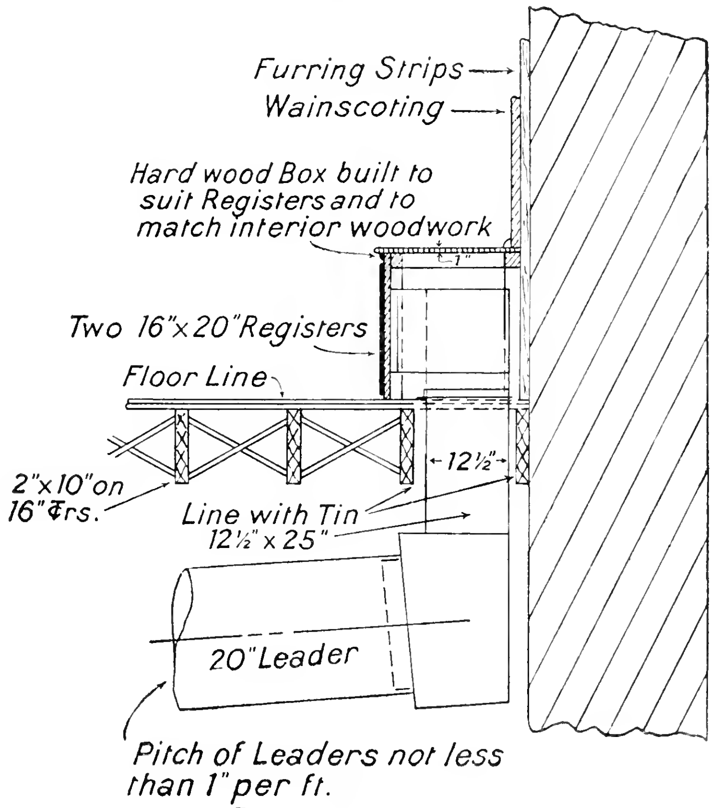


FIG. 33.

The weight of water to be added is then,

$$8{,}700 \times 8.069 \times 0.50 - 8{,}100 \times 2.124 \times 0.50 = 26{,}460 \text{ grains.}$$

$$26{,}460 \,/\, 7{,}000 = 3.78 \text{ lb.}$$

The heat required to evaporate this water at 70° is obtained by reference to the table of Properties of Air and Vapor Mixtures, and at 70° = $16.61 \times 0.50 \left(1 - \frac{2.128}{8.071}\right) = 6.12$ B.t.u. per 1 lb. of air.

And since $8{,}700 \times 0.075 = 652$ lb. then $652 \times 6.12 = 3{,}990$ B.t.u. per hour, or since the latent heat at 70° = 1,052 B.t.u., we may take $3.78 \times 1{,}052 = 3{,}980$ B.t.u. The heat necessary to raise the vapor added to the register temperature is all returned to the system when the vapor cools down to 70°, so that no account has been taken of same.

This problem can also be solved very readily by reference to the Psychrometric Chart. See chapter on "Air Conditioning."

Example. (Case 2.) Air at 70° can hold 0.0156 lb. vapor per 1 lb. of air and at 32°, 0.0038 lb. of vapor per 1 lb. of air. Hence water vapor required $= 652 \times (0.0156 - 0.0038) \times 0.5 = 3.82$ lb. per 10,000 cu. ft. entering at 150° per hour.

Using the average value of 3.80 lb. and assuming a 24-hr. day and a supply of 50,000 cu. ft. per hour at registers we have for an average house:

$$\frac{5 \times 3.80 \times 24}{8.33} = 54.75 \text{ gals. of water per day.}$$

This will require for the temperatures stated above, $5 \times 3{,}990 \times 24 / 8{,}000 = 60$ lb. of coal per 24 hours, based on the assumption of continuous service and a 12,000 B.t.u. coal yielding 66 2/3 % of its heat value to the water.

Humidifiers. It is readily apparent that no ordinary *water pan,* as usually installed in the lower part of a warm air furnace casing, will supply even a small fraction of the water vapor required according to the above calculation.

Special evaporating tanks at the level of the fire-pot (Fig. 34) or water backs which are connected to a constant source of water supply kept under ball cock control, are sometimes employed. The fluctuations in fire intensity, and condition and temperature of the outside air make it almost impossible to proportion these tanks so as to maintain the proper rate of evaporation. In fact, it is doubtful if entirely satisfactory results can be obtained without some sort of automatic control such as could be maintained by means of a humidostat operating on a live-steam supply line. See chapter on "Temperature and Humidity Control."

WARM-AIR RADIATOR SYSTEM OF HEATING

Essential Features of the System. A modified system of furnace heating using warm-air radiators has been developed by the *International Radiator Co.* for use with a double system of stacks and leaders in which all the air is recirculated between the furnace and pressed steel radiators placed in each room. The system operates on exactly the same principle as a hot water system, which is due to the difference in density of the warm flow and the cooler return column of air.

The air used for heating does not pass into and mix with the air of the room at all, but is confined in the radiator, which gives off its heat by radiation and convection similarly to a direct steam or hot-water radiator at the same temperature.

Equipment Required. The *furnace* is selected as for any ordinary furnace system, basing grate area and heating surface on the total heat loss from the building, plus 20 to 25% for transmission losses, and the chimney flue and smoke connection are proportioned according to height of chimney and grate size. Furnace casing and top must be covered with at least 1″ of plastic asbestos or asbestos blocks.

The cold-air duct is *omitted* entirely in this system, and in its place the *return leaders* are connected to the bottom of the furnace for recirculation of the cooled air coming from the radiators. No water-pan or humidifier is placed within the furnace casing with this system.

The *leaders and stacks* are proportioned as for the ordinary gravity system, but, of course, must be run on the two-pipe principle, with return stacks and leaders of the same size as flow lines. The use of trunk leaders is common in this work, thereby reducing the number of basement pipes very materially. All leaders and stacks should be thoroughly insulated against heat loss with at least ¼″ to ½″ of asbestos air cell covering since the average temperatures maintained

in the radiators are as high as 185° in zero weather. The heat loss from unprotected piping of large diameter would be very severe at this temperature, and impose an unusual tax on the furnace in cold weather.

The *radiators* (Fig. 35) are of No. 28 U. S. S. gage pressed steel, made up in two sizes of

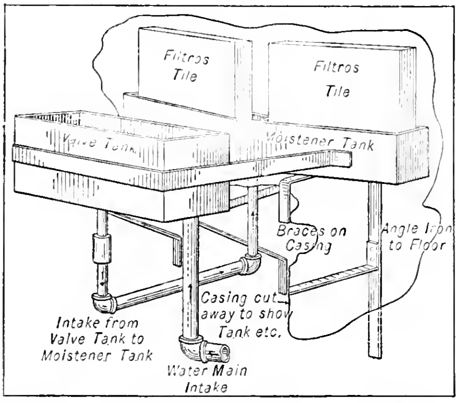


FIG. 34. EVAPORATING TANK AND CONNECTIONS FOR ATTACHMENT TO FURNACE CASING.

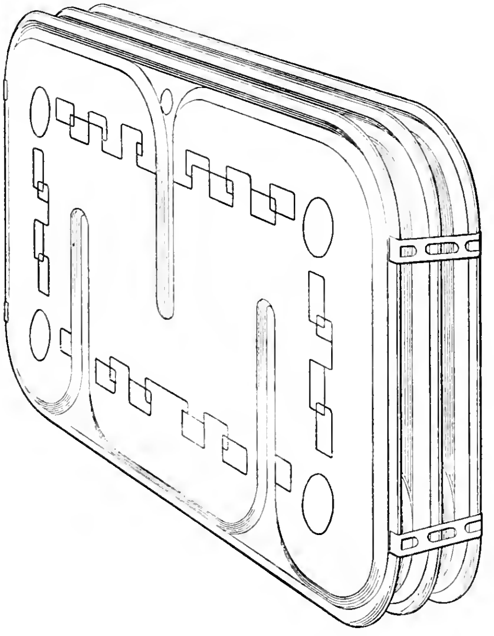


FIG. 35. THREE SECTION INTERNATIONAL RADIATOR.

sections. Size (A) sections are 24″ x 42″ with 7″ collar connections and of 14 sq. ft. effective heating surface per section. Size (B) sections are 24″ x 24″ with 7″ collar connections and of

8 sq. ft. effective heating surface. Either size of section may be assembled into radiators of 2, 3, 4 or 5 sections, and the size of radiator for a given room readily determined as follows:

$$A = \frac{H}{K \times (t_2 - t_1)}, \text{ where}$$

A = area of surface to be installed in sq. ft.
H = B.t.u. loss from the room per hour.
K = coefficient of transmission = 1.65.
t_2 = average temp. of air in rad. = 185°.
t_1 = room temp. desired = 70° usually.

These warm air radiators are mounted on brackets on an interior wall, containing the flow and return stacks, and placed just above the baseboard. Each radiator has a butterfly control damper in the return collar. The radiators are therefore off the floor and only extend from 4″ to 6″ into the room. All radiators are enamelled to suit the decorative requirements, and are exceptionally sanitary in their design, and readily cleaned. There are no pipes coming through the floor and no cutting of carpets is necessary.

Ventilation. Ventilation may be secured by open fireplaces or by suitable vent flues and vent registers if a number of occupants must be provided for. The presence or absence of a ventilating system will not affect the operation of this heating system except that larger radiators will be necessary to provide for the greater heat loss due to the increased number of air changes resulting from the installation of a ventilating flue or system.

Advantages of the System. The advantages of this system of heating are found in the absence of water, steam or vapor which may leak and cause damage, and in the absence of coal dust and furnace gases in the heated rooms, since all the air passing the furnace is confined in a tight system at atmospheric pressure. A wide range in the temperature of the radiating surfaces is possible and readily attainable, depending on the intensity of the fire in the furnace, which should be varied to suit outside weather conditions. This is a great advantage, and results in the same economy in operation as for a hot-water system.

The system responds readily and circulation starts immediately after the fire is lighted. The system requires no draining or filling with water when shutting down or starting up at the beginning or end of the season, and there is no possibility of freezing at any time. The cost of installation is about 75% of that for a steam system, and 60% of that for hot water heating.

CHAPTER XV

HOT BLAST HEATING

GENERAL FEATURES

The mechanical indirect method of heating, commonly known as the blower or hot blast system, particularly adapted to the warming and ventilating of large structures, is made up of three units: (1) A heater constructed of pipes, tubes, or cast-iron sections, through which steam, hot water or hot gas may be passed. (2) A fan or blower to circulate air over the heater surfaces; the air acting as a heat carrier or medium of heat transfer. (3) A system of ducts or pipes to convey the heated air from the heater to points where heat may be required.

When the heater is located between the fan and main duct, the combination is termed *blow through*, and when the fan is installed between the heater and the duct, the arrangement is known as *draw through*. These two arrangements are shown by Fig. 8. The *draw through* combination is more often used for shop and factory installations where compactness is desirable, the *blow through* combination being used principally for "hot and cold" systems as installed in schools and public buildings.

Advantages of this System. The advantages of the blower or hot blast system over that of direct radiation briefly summarized are:

I. When ventilation is a requirement in order to maintain a healthful atmosphere, this method affords a positive means of accomplishing this particularly desirable result, which is entirely independent of the changing climatic conditions.

II. When a standard humidity of the air is to be maintained, a feature which is becoming to be more generally recognized as desirable in any heating and ventilating installation, and quite essential to the successful manufacture of some materials, the humidifying apparatus may readily be made an integral part of the system.

III. A much smaller amount of radiating surface is required to perform an equal heating duty, with a consequent reduction in the number of steam-tight joints, unions and valves to keep in repair.

IV. The air leakage being mostly outward, the building will in general be freer from drafts and more uniformly heated. If the air is simply re-circulated, no fresh air being taken into the heating system from the outside, the above statement does not apply. The pressure of the air in the building, even when all of the air is taken into the heating system from the outside, is comparatively feeble, and some air will enter by infiltration through the window and door cracks on the windward side of the building, although the statement is often made that the leakage being all outward, prevents the infiltration of cold air from the outside.

V. This system is more easily regulated, and readily responds to changing outside temperatures.

VI. The air entering for ventilation may be conveniently cooled in summer, either by the circulation through the heater of cold water or of brine previously cooled by mechanical refrigeration.

Simply running the fan will in itself greatly relieve the oppressiveness in hot sultry weather. And when cold water is circulated through the coils the difference is very noticeable.

Typical Arrangements. Several arrangements for the heating of industrial plants by the hot blast system are shown by Figs. 1 to 5. When ventilation is not a requirement or relatively unimportant as is frequently the case in shop or factory heating when the number of persons vitiating the air is small compared with the cubical contents of the building, the air may

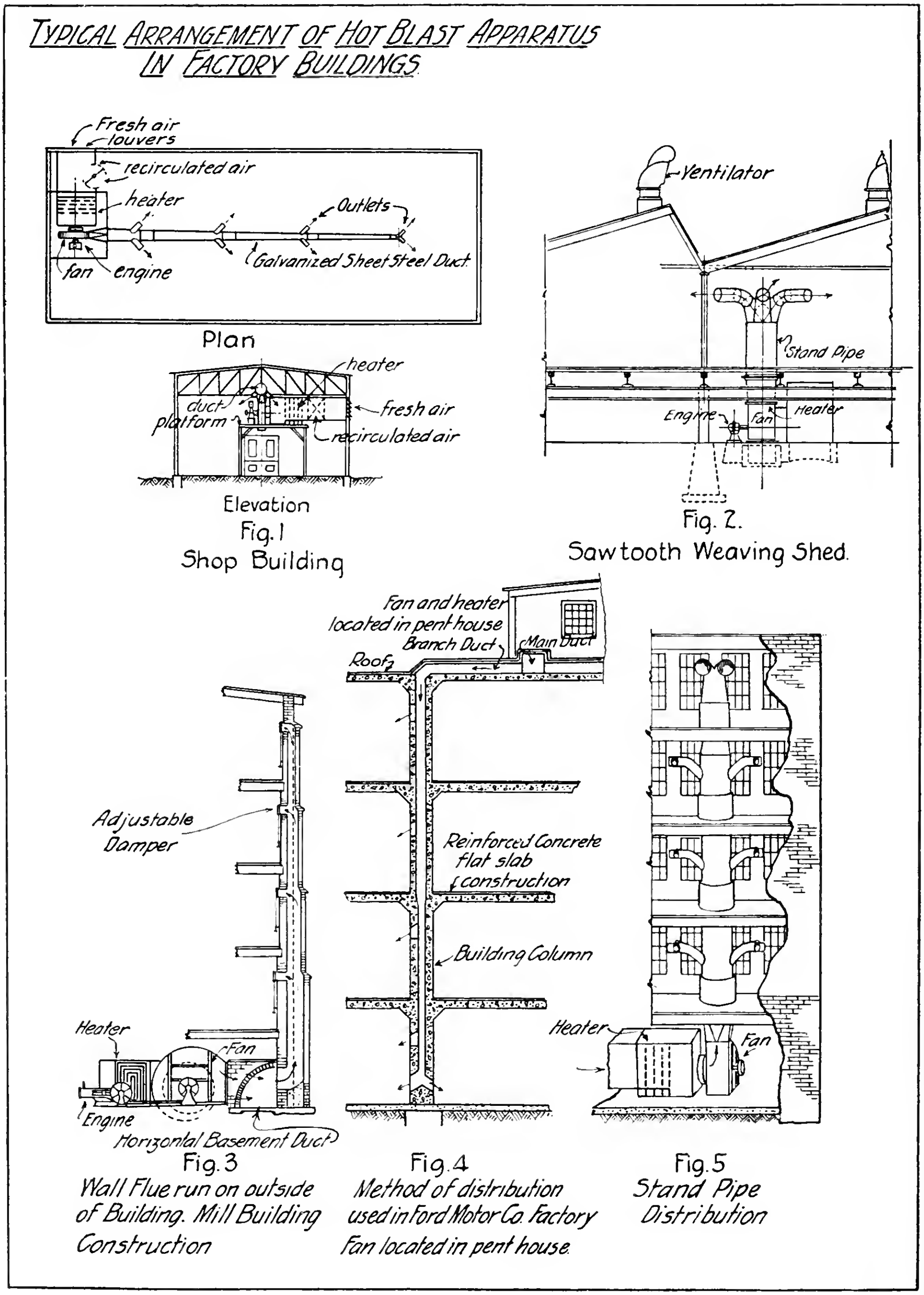


Fig. 1
Shop Building

Fig. 2.
Sawtooth Weaving Shed.

Fig. 3
Wall Flue run on outside of Building. Mill Building Construction

Fig. 4
Method of distribution used in Ford Motor Co. Factory Fan located in pent house.

Fig. 5
Stand Pipe Distribution

be simply recirculated, sufficient fresh air for ventilation being supplied by infiltration. The amount of heat to be supplied the heater in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation considerable economy may be effected by recirculating a portion of the air.

In this case only sufficient fresh air is drawn into the system from the outside to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated.

This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as shown.

If the fresh air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Amount of Air to be Circulated for Heating. The weight of air to be circulated per hour for heating a room or building is found by dividing the heat loss (H) by the amounts of heat given up by one pound of air in cooling from the temperature at the duct outlets to the mean room temperature.

Let H = heat loss of room, B.t.u. per hour.
= $(a + b - c)$ see chapter on "Heat Loss from Buildings."
M = weight of air to be introduced in room per hour.
t = mean inside temperature.
t_d = Temperature of air leaving duct outlets.

Then $$M = \frac{H}{0.24(t_d - t)} \qquad (1)$$

The effect of vapor on the weight of air is so slight that it may be safely neglected in these calculations. For problems in which humidity plays an important part see chapter on "Air Conditioning, Air Washers," etc.

The temperature t_d depends upon the temperature of the air entering the heater, the velocity through the clear area, the amount of heating surface and temperature of the steam. This temperature in practice ordinarily ranges from 125° to 150° F. and may be readily determined for any specified condition by the data given later under "Heaters." The temperature of the air leaving the duct outlets for ordinary installations when the ducts are not run underground or in outside walls, may be assumed the same as the temperature (t_2) of the air leaving the heater. Any loss in temperature in this case goes toward heating the building and is therefore not a direct loss.

If, however, the ducts are run underground or in outside walls a considerable loss in temperature may occur, which is a direct loss, and must be provided for by *increasing the temperature of the air leaving the heater* by an amount equal to the estimated temperature drop in the ducts.

Temperature of Air Entering Heater.

Let t_1 = temperature of air entering heater.
t_0 = outside temperature.
t = mean inside temperature.
t_2 = temperature air leaving heater.

(*a*) When the air is all recirculated $t_1 = t$.

(*b*) When fresh air only is circulated $t_1 = t_0$.

(*c*) When a portion of the air is recirculated the resulting temperature of the mixture of fresh and recirculated air may be found by the method of mixtures.

Let M_v = weight of fresh air, lb. required per hour for ventilation.
= $0.075 \times 1{,}800 \times$ no. persons (usual requirement).
M_r = weight of air that may be recirculated.

$M = M_v + M_r$.

$0.24\ (M_v + M_r)\ (t_d - t) = H$. Having assumed or fixed the value of t_d the only unknown quantity is M_r.

$$M_r = \frac{H}{0.24\ (t_d - t)} - M_v \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The temperature t_1 may then be found as follows:

$$\begin{aligned} M_v \times (t_0 + 460) &= A \\ M_r \times (t + 460) &= B \\ (M_v + M_r)\ (t_1 + 460) &= A + B \end{aligned}$$

$$\text{or } t_1 = \frac{A + B}{M_v + M_r} - 460$$

Example. The heat loss H for a certain factory building is 70,600 B.t.u. per hour. The number of men employed is 50. Mean inside temperature $t = 65°$ F. Outside temperature $t_0 = 0°$ F. Ventilation is to be provided at the rate of 1800 cu. ft. of fresh air per hour per person. Assumed temperature of air leaving duct outlets is 135° F.

$M_v = 0.075 \times 1800 \times 50 = 6750$ lb. per hour fresh air for ventilation. The weight of air that may be recirculated is $M_r = \dfrac{706{,}000}{0.24(135 - 65)} - 6750 = 35{,}273$ lb. per hour.

The temperature of the air entering the heater will be:

$$\begin{aligned} 6750 \times (\ 0 + 460) &= \ \ 3{,}105{,}000 \\ 35{,}273 \times (65 + 460) &= 18{,}516{,}750 \end{aligned}$$

$$t_1 = \frac{21{,}621{,}750}{42{,}023} - 460 = 55°\text{ F.}$$

If *fresh air only* is to be used, as in school-house and public building heating, the weight of air to be circulated is determined directly by the ventilation requirement.

$$\text{Then } M = M_v = \frac{H}{0.24\ (t_d - t)} \text{ or } t_d = \frac{H + 0.24\ M_v t}{0.24\ M_v} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Temperature of Air at Duct Outlets. When heating a building by the hot-blast system containing a number of rooms having different heat losses and ventilation requirements, it is obviously impossible to maintain the desired temperature by controlling the temperature (t_2) of the air leaving the heater at one point.

The temperature t_d of the air leaving the duct outlets will ordinarily be different for each room in the building as shown by the following example. This result is accomplished by the double plenum chamber system later described.

Example. Let it be required to determine the temperature of the entering air (t_d) to offset the heat loss and provide ventilation for the several rooms as given by the following table. Inside temperature (t) to be maintained is 70°.

TABLE 1

Room Number	No. of Occupants n	Ventilation: Cu. Ft. per Hour at 70° 1,800 x n	Ventilation: Weight per Hour M_v	Heat Loss H	Temperature of Entering Air t_d Formula (3)
A-1	50	90,000	6750	32,000	89.5
A-2	53	95,400	7125	21,000	82.2
Hall	..	30,000	2250	4,000	77.4

HOT BLAST HEATING

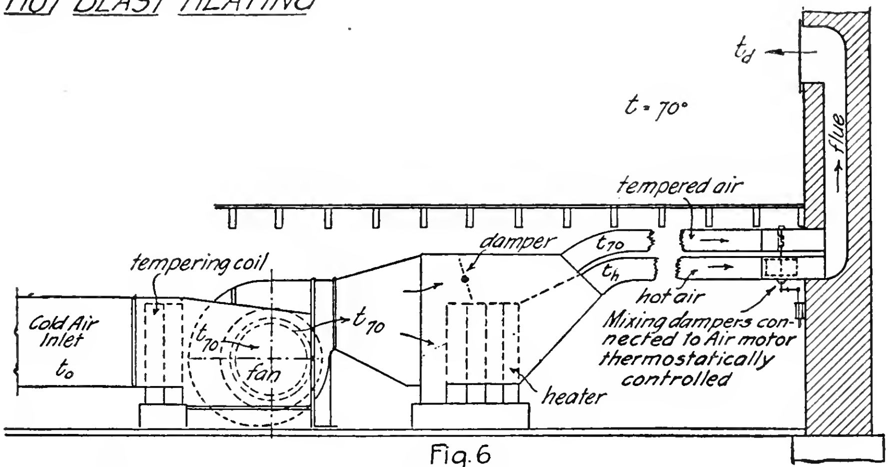


Fig. 6
Double Duct System

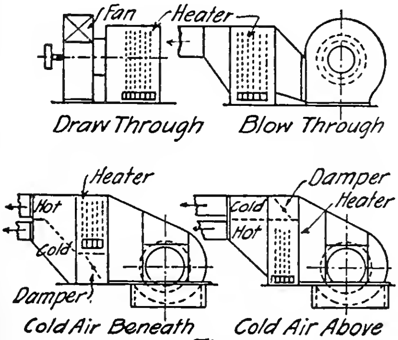


Fig. 8
Types of Heater Jackets

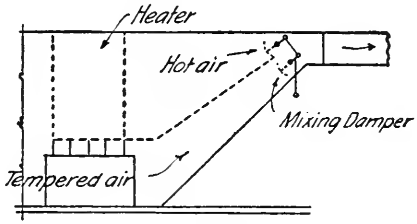


Fig 7
Single Duct System

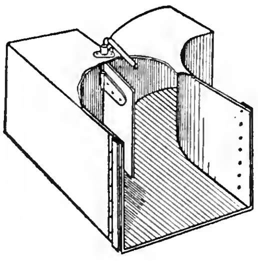


Fig. 9
Deflecting Damper

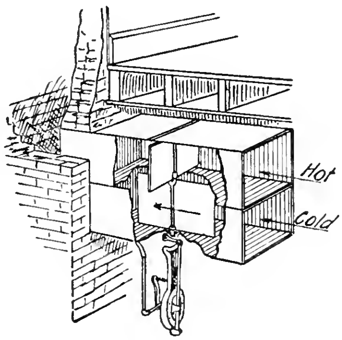


Fig. 10
Thermostatically Controled
Mixing Damper

Temperature of Air Leaving Heater. If all of the air is first warmed by the *tempering coil* to 70° F. and a mixture of approximately $(1 - x)$ parts tempered air and x parts hot air is to be used, then the required temperature of the hot air leaving the heater may be determined, for any particular case, by the method of mixtures previously given, or assuming this temperature, the proportions of hot and tempered air may be determined.

Example. Required the temperature of the hot air (t_h) leaving the heater for Room A – 2, (Table 1) if the mixture entering the room is made up of ½ tempered air at 70° and ½ hot air. Total weight of air entering room is 7125 pounds per hour or 3562.5 pounds of tempered air and 3562.5 pounds of hot air.

$3562.5 \times (70 + 460) + 3562.5 \times (t_h + 460) = 7125. \times (82.2 + 460) \therefore t_h = 94.$

Assuming a temperature of $t_h = 120°$ determine the relative proportions, by weight, of the mixture required. Let x = parts of hot air in mixture, then $(1 - x)$ = parts of tempered air.

$$x\,(120 + 460) + (1 - x) \times (70 + 460) = (82.2 + 460)$$

Then $x = 0.244$ and $(1 - x) = 0.756$.

Air Supplied for Ventilating Purposes Only. A combination of direct radiation, to offset the heat loss H, and a hot blast system, to supply the fresh air needed for ventilation, is sometimes installed.

In this case it is customary to install a heater of sufficient capacity to warm the air for ventilation to about 80°. The heater used for this purpose is made three sections deep.

Hot and Cold Systems. In order to accomplish the results required in the preceding example, the so-called *hot and cold* or *double plenum chamber* system is used.

All of the air drawn into the system from the outside is first passed through a *tempering coil* which is designed to heat the air to approximately 70°. A portion of the tempered air is then passed through a heater and raised to 125° to 150°. Then if varying proportions of the hot and tempered air are correctly mixed the resulting temperature (t_d) is readily controlled without varying the quantity of air discharged, which evidently must remain constant on account of the ventilation requirement. There are two methods of distribution used as shown by Figs. 6 and 7. Referring to Fig. 7 it will be seen that the hot and tempered air meet at the end of the plenum chamber at the entrance to the ducts, and the temperature of the mixture is controlled by the mixing dampers which may either be hand operated or placed under automatic thermostatic control. It will be observed that the plenum chamber is divided, and that each duct serving a room has its own independent set of mixing dampers.

This method of distribution is known as the *single duct* system and is frequently employed where the installation of the *double duct* system, as described below, is not feasible or is undesirable.

Fig. 6 shows a double set of ducts run from the plenum chamber to the base of each vertical flue, one carrying the hot air and the other the tempered air, the mixing being done at the base of the flue as shown. The mixing dampers (Fig. 10) may be controlled by hand by means of a chain carried up the flue and run into the room at a point several feet above the floor line or placed under automatic thermostatic control through the medium of a compressed air operated damper. See "Temperature and Humidity Control."

HOT BLAST HEATERS

Pipe Coil Heaters. The type of heater that has been a standard for a number of years, for hot blast work, is made up of four or eight vertical rows, depending on the manufacturer, of 1-inch pipe screwed into a cast-iron base and spaced on 2⅛" to 2¾" centers, the pipes in each row being cross connected at the top by the use of nipples and ells. Figs. 11 and 12 show two different arrangements used. The arrangement of coils and the method of dividing the base by a vertical partition, running longitudinally, in order to separate the supply and return as used by the *American Blower Co.* are shown by Fig. 11.

A base with its accompanying pipes is termed a *section*.

The heater is made up of a number of such sections enclosed by a *sheet steel jacket*, usually

No. 22 or 24 U. S. S. gage, which is connected to the fan either on the suction or discharge side. See types of heater jackets, Fig. 8, and dimensions of casings for pipe coil heaters, Fig. 16.

Fig. 54 shows *a complete assembly* of this type of heater. The first section is supplied by the exhaust from the fan engine while the last three sections are supplied with live steam through the 4″ cast-iron header shown. The base of each section is dripped on both sides of the

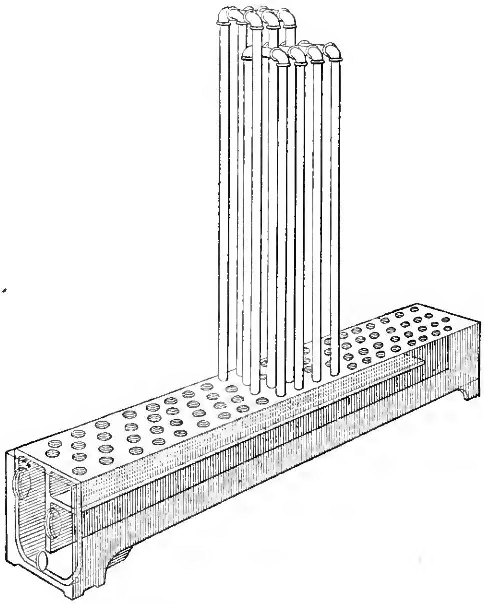


FIG. 11. PIPE COIL HEATER BASE.
(*American Blower Co.*)

partition, the supply side being connected into the main return through a water sealed drip loop to prevent blowing live steam into the main return, which is connected to a steam trap in the usual manner.

Owing to the difference in pressure existing in the first section, supplied with exhaust steam, and the remaining sections supplied with live steam, the return from the first section must be independently trapped. When the entire heater is to be supplied by exhaust steam from other sources, then all sections may be connected to the same return and trap.

The base and piping arrangement used by the *B. F. Sturtevant Co.* and the *Buffalo Forge Co.* is shown by Fig. 12 and also by Figs. 14 and 15; in this case the base is divided by a cross partition at the center, to separate the supply and return for the arrangement of pipes used.

Heaters may be constructed with the pipe coils perpendicular to the direction of air flow (Fig. 14) in which case they are known as *Regular Open Area Pattern Heaters (O. A. P.)*, or they may have the pipe coils parallel to the direction of air flow (Fig. 15) in which case they are known as *Return Bend Heaters (R. B.)*.

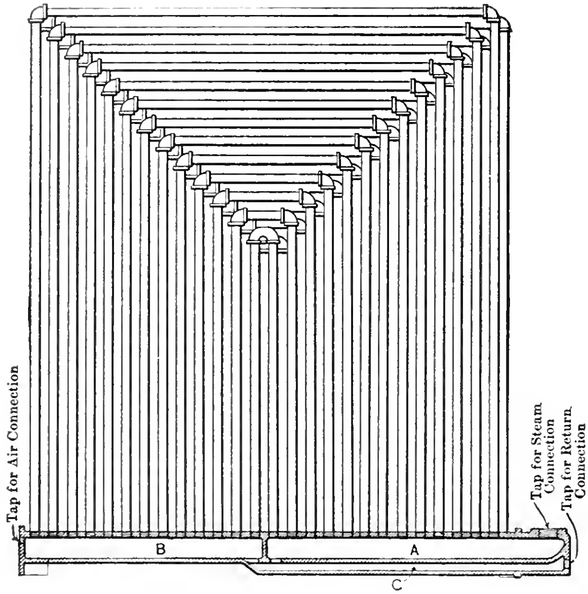


FIG. 12. SECTIONAL VIEW—OPEN AREA PATTERN HEATER.

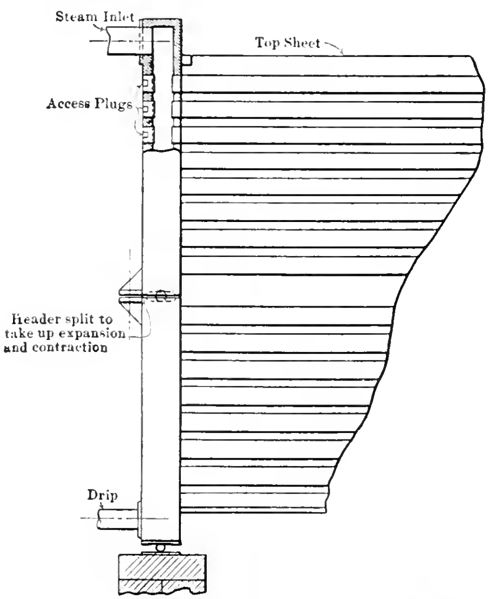


FIG. 13. GREEN'S BOX-HEADER-TYPE HOT-BLAST HEATER COIL, SHOWING ACCESS PLUGS FOR GETTING AT TUBES AND ROLLERS SUPPORTING TWO HALVES OF FRONT HEADER TO ALLOW FOR EXPANSION AND CONTRACTION.

Sizes and Dimensions of Pipe Coil Heaters. The following tables give the sizes and dimensions of the *Buffalo Forge Co.'s* standard heaters and casings:

TABLE 2

SIZES AND DIMENSIONS OF *BUFFALO FORGE CO.* STANDARD HEATERS

Manner of Piping	Number of Pipes	Length of Section	Section Number	Extreme Height Section	Width of Section	Lin. Feet of 1-Inch Pipe per Section	Total Effective Sq. Ft. Heating Surface	Equivalent in Lin. Feet of 1-Inch Pipe	Clear Area for Air Passage Sq. Ft.	Weight
R.O.A.......	56	3′ 4 row	1 A	3′- 4″	8½″	140	54.7	159	4.4	473
			2 A	3′-10″	8½″	168	64.2	186	5.2	515
			3 A	4′- 4″	8½″	196	74.0	215	6.0	565
			4 A	4′-10″	8½″	224	83.7	243	6.8	616
			5 A	5′- 4″	8½″	252	93.3	271	7.6	656
			6 A	5′-10″	8½″	280	102.5	298	8.4	708
R.O.A.......	72	4′ 4 row	1 B	5′- 4″	8½″	320	119.0	346	9.7	819
			2 B	5′-10″	8½″	356	131.5	382	10.7	877
			3 B	6′- 4″	8½″	392	143.9	418	11.2	938
			4 B	6′-10″	8½″	428	156.5	455	12.6	1003
R.O.A.......	80	4′-6″ 4 row	1 C	5′-10″	8½″	396	148.2	431	12.1	997
			2 C	6′- 4″	8½″	436	162.0	480	13.1	1055
			3 C	6′-10″	8½″	476	174.8	507	14.2	1127
			4 C	7′- 4″	8½″	516	188.6	548	15.3	1174
R.O.A.......	88	5′ 4 row	1 D	6′- 4″	8½″	476	174.3	507	14.1	1182
			2 D	6′-10″	8½″	520	189.3	550	15.4	1262
			3 D	7′- 4″	8½″	564	204.8	595	16.6	1325
			4 D	7′-10″	8½″	608	219.8	638	17.7	1407
R.O.A.......	104	6′ 4 row	1 E	7′- 4″	8½″	674	245.0	712	19.8	1505
			2 E	7′-10″	8½″	726	262.9	763	21.3	1600
			3 E	8′- 4″	8½″	778	280.8	816	22.7	1695
			4 E	8′-10″	8½″	830	298.7	868	24.2	1770
R.O.A.......	64	7′ 2 row	1 F	8′- 4″	6″	477	173.1	503	28.1	1198
			2 F	8′-10″	6″	509	184.3	535	30.0	1244
			3 F	9′- 4″	6″	541	195.3	567	31.7	1303
			4 F	9′-10″	6″	573	205.3	596	33.3	1350
R.B.........	128	7′ 4 row	1 G	7′- 4″	8½″	796	291.0	845	23.6	1845
			2 G	7′-10″	8½″	860	313.2	910	25.4	1950
			3 G	8′- 4″	8½″	924	335.2	974	27.2	2055
			4 G	8′-10″	8½″	988	357.2	1037	29.0	2160
			5 G	9′- 4″	8½″	1052	379.2	1101	30.7	2280
			6 G	9′-10″	8½″	1116	401.2	1163	32.5	2380
R.B.........	154	8′-6″ 4 row	1 H	8′- 4″	10″	1119	410.2	1190	33.2	2675
			2 H	8′-10″	10″	1196	436.8	1265	35.3	2800
			3 H	9′- 4″	10″	1273	463.5	1345	37.6	3075
			4 H	9′-10″	10″	1350	490.0	1421	39.8	3200
			5 H	10′- 4″	10″	1427	516.6	1499	41.8	3325
			6 H	10′-10″	10″	1504	543.2	1578	44.0	3455
R.B.........	170	9′-6″ 4 row	1 I	8′- 4″	10″	1231	452.3	1313	36.7	3205
			2 I	8′-10″	10″	1316	481.6	1396	39.0	3350
			3 I	9′- 4″	10″	1401	510.9	1481	41.4	3485
			4 I	9′-10″	10″	1486	540.2	1570	43.8	3625
			5 I	10″- 4″	10″	1571	569.5	1651	46.0	3770
			6 I	10′-10″	10″	1656	598.7	1739	48.4	3910
			7 I	11′- 4″	10″	1741	628.0	1821	50.8	4060
			8 I	11′-10″	10″	1826	657.3	1910	53.2	4200

Reg. O.A.P. Heater
(Buffalo Forge Co.)

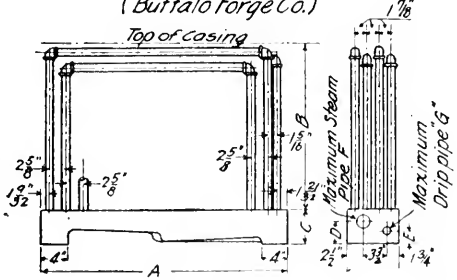


Fig. 14

Return Bend Heater
(Buffalo Forge Co.)

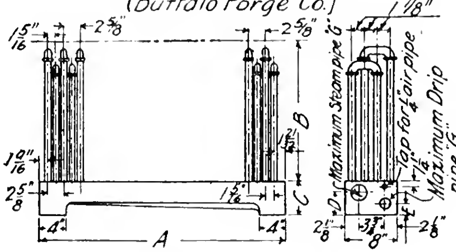
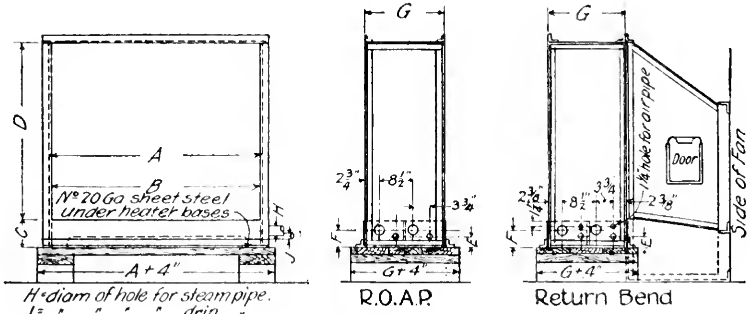


Fig. 15

(Figs. 14 & 15)
H = Number of pipe in section.
J = Clear area in sq. ft.
K = Lineal feet of 1" pipe
See table 3 for dimensions

Dimensions of Heater Case for Buffalo Standard Heater.



H = diam. of hole for steam pipe.
J = " " " " drip "

Fig. 16

NOTE:—Connection as shown in full lines for full Housing Fans up to and including 120". Connection as shown in dotted lines for full Housing Fans over 120" and all ¾ Housing Fans.

SIZE OF SECTION LENGTH	HEIGHT	A	B	C	D	E	F	G	H 0 lbs	H 5 lbs	H 20 lbs	H 60 lbs	J
3 ft.	3'-4"	38¾	38⅜	5	34	2	3⅛	8½ × No. of Section	2⅛	1⅞	1½	1½	1½
	3'-10"	"	"	"	38	"	"		"	"	"	"	"
	4'-4"	"	"	"	44	"	"		2⅝	2⅛	1⅞	"	"
	4'-10"	"	"	"	50	"	"		"	"	"	"	"
	5'-4"	"	"	"	56	"	"		"	2⅝	"	"	"
	5'-10"	"	"	"	62	"	"		"	"	2⅛	"	"
4 ft	5'-4"	49¼	48⅞	5⅝	56	2⅛	3½		3⅛	2⅝	2⅛	1½	1½
	5'-10"	"	"	"	62	"	"		"	"	"	1⅞	"
	6'-4"	"	"	"	68	"	"		"	"	"	"	"
	6'-10"	"	"	"	74	"	"		"	3⅛	2⅝	"	1⅞
4 ft-6 in	5'-10"	54½	54⅛	5⅝	62	2⅛	3½		3⅛	2⅝	2⅛	1⅞	1½
	6'-4"	"	"	"	68	"	"		"	3⅛	2⅝	"	1⅞
	6'-10"	"	"	"	74	"	"		"	"	"	"	"
	7'-4"	"	"	"	80	"	"		"	"	"	"	"
5 ft	6'-4"	59¾	59⅜	6⅜	68	2 5/16	3 5/16		3⅛	3⅛	2⅝	1⅞	1⅞
	6'-10"	"	"	"	74	"	"		3¾	"	"	"	"
	7'-4"	"	"	"	80	"	"		"	"	"	2⅛	"
	7'-10"	"	"	"	86	"	"		"	"	"	"	"
6 ft	7'-4"	70¼	69⅞	6⅜	80	2 5/16	3 5/16		3¾	3¾	2⅝	2⅛	2⅛
	7'-10"	"	"	"	86	"	"		"	"	3⅛	"	"
	8'-4"	"	"	"	92	"	"		"	"	"	"	"
	8'-10"	"	"	"	98	"	"		"	"	"	2⅝	"
7 ft.	8'-4"	86	85⅝	6⅜	92	2 5/16	3 5/16		3¾	3¾	3⅛	2⅝	2⅛
	8'-10"	"	"	"	98	"	"		"	"	"	"	"
	9'-4"	"	"	"	104	"	"		"	"	"	"	"
	9'-10"	"	"	"	110	"	"		"	"	"	"	"

TABLE 3

DIMENSIONS OF REG. O. A. P. AND RETURN BEND HEATERS

(Buffalo Forge Co.)

Size of Section		A	B	C	D	E	F	G	H	J	K
Length	Height										
3 ft.	3'- 4"	38 3/8"	34"	5"	3 1/8"	2"	2"	1"	56	5.1	158
	3'-10"	38 3/8"	38"	5"	3 1/8"	2"	2"	1"	56	5.4	178
	4'- 4"	38 3/8"	44"	5"	3 1/8"	2"	2"	1"	56	6.1	193
	4'-10"	38 3/8"	50"	5"	3 1/8"	2"	2"	1"	56	6.9	221
	5'- 4"	38 3/8"	56"	5"	3 1/8"	2"	2"	1"	56	7.7	249
	5'-10"	38 3/8"	62"	5"	3 1/8"	2"	2"	1"	56	8.5	277
4 ft.	5'- 4"	48 7/8"	56"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	72	9.8	320
	5'-10"	48 7/8"	62"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	72	10.8	356
	6'- 4"	48 7/8"	68"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	72	11.8	392
	6'-10"	48 7/8"	74"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	72	12.9	428
4 ft. 6 in.	5'-10"	54 1/8"	62"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	80	12.0	396
	6'- 4"	54 1/8"	68"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	80	13.0	436
	6'-10"	54 1/8"	74"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/8"	80	14.0	477
	7'- 4"	54 1/8"	80"	5 5/8"	3 1/2"	2 1/8"	2 1/2"	1 1/4"	80	15.0	516
5 ft.	6'- 4"	59 3/8"	68"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	88	14.3	479
	6'-10"	59 3/8"	74"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	88	15.6	523
	7'- 4"	59 3/8"	80"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	88	16.8	567
	7'-10"	59 3/8"	86"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	88	17.8	611
6 ft.	7'- 4"	69 7/8"	80"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	104	19.7	670
	7'-10"	69 7/8"	86"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	104	21.2	722
	8'- 4"	68 7/8"	92"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	104	22.7	774
	8'-10"	69 7/8"	98"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	104	24.2	826
7 ft.	8'- 4"	85 5/8"	92"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	128	27.0	960
	8'-10"	85 5/8"	98"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	128	29.0	1024
	9'- 4"	85 5/8"	104"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	128	30.8	1088
	9'-10"	85 5/8"	110"	6 3/8"	3 15/16"	2 5/16"	3"	1 1/2"	128	32.5	1152

NOTE.—See Figs. 14–16 for sketch of Heaters and Casings.

TABLE 4

STEAM, DRIP AND AIR CONNECTIONS FOR REG. O. A. P. HEATERS

(Buffalo Forge Co.)

Size of Heater	Size of Steam Supply				Size of Drip	Size of Main Drip				
	0 Lbs.	5 Lbs.	20 Lbs.	60 Lbs.		2 Sect.	3 Sect.	4 Sect.	5 Sect.	6 Sect.
3'-0" x 3'- 1"	1 1/2	1 1/4	1	1	1	1 1/4	1 1/2	1 1/2	2	2
3'-0" x 3'-10"	1 1/2	1 1/4	1	1	1	1 1/4	1 1/2	1 1/2	2	2
3'-0" x 4'- 4"	2	1 1/2	1 1/4	1	1	1 1/4	1 1/2	1 1/2	2	2
3'-0" x 4'-10"	2	1 1/2	1 1/4	1	1	1 1/4	1 1/2	1 1/2	2	2
3'-0" x 5'- 4"	2	2	1 1/4	1	1	1 1/4	1 1/2	1 1/2	2	2
3'-0" x 5'-10"	2	2	1 1/2	1	1	1 1/4	1 1/2	1 1/2	2	2
4'-0" x 5'- 4"	2 1/2	2	1 1/2	1	1	1 1/4	1 1/2	2	2	2 1/2
4'-0" x 5'-10"	2 1/2	2	1 1/2	1 1/4	1	1 1/4	1 1/2	2	2	2 1/2
4'-0" x 6'- 4"	2 1/2	2	1 1/2	1 1/4	1	1 1/4	1 1/2	2	2	2 1/2
4'-0" x 6'-10"	2 1/2	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
4'-6" x 5'-10"	2 1/2	2	1 1/2	1 1/4	1	1 1/4	1 1/2	2	2	2 1/2
4'-6" x 6'- 4"	2 1/2	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
4'-6" x 6'-10"	2 1/2	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
4'-6" x 7'- 4"	2 1/2	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
5'-0" x 6'- 4"	2 1/2	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
5'-0" x 6'-10"	3	2 1/2	2	1 1/4	1 1/4	1 1/2	2	2	2 1/2	2 1/2
5'-0" x 7'- 4"	3	2 1/2	2	1 1/2	1 1/4	1 1/2	2	2	2 1/2	2 1/2
5'-0" x 7'-10"	3	2 1/2	2	1 1/2	1 1/4	1 1/2	2	2	2 1/2	2 1/2
6'-0" x 7'- 4"	3	3	2	1 1/2	1 1/2	2	2 1/2	2 1/2	3	3
6'-0" x 7'-10"	3	3	2 1/2	1 1/2	1 1/2	2	2 1/2	2 1/2	3	3
6'-0" x 8'- 4"	3	3	2 1/2	1 1/2	1 1/2	2	2 1/2	2 1/2	3	3
7'-0" x 8'-10"	3	3	2 1/2	2	1 1/2	2	2 1/2	2 1/2	3	3

NOTE.—See Figs. 14 and 16.

TABLE 5

HOT BLAST HEATERS

(B. F. Sturtevant Co.)

Size of Section	Nominal Height of Pipes Ft.	In.	Two Row Lineal Ft. 1-In. Pipe	Two Row Total Area in Sq. Ft.	Four Row Lineal Ft. 1-In. Pipe	Four Row Total Area in Sq. Ft.	Number of Pipes 2 Rows	Width of Jacket	Free Area Through Section Sq. Ft.
3 Feet	3	0	97	39.5	194	76.5	34	37¾	3.85
	3	6	115	45.5	230	88.5	..	....	4.52
	4	0	133	52.0	266	101.0	..	....	5.21
	4	6	148	57.0	296	111.5	..	....	5.76
	5	0	166	63.5	332	124.0	..	....	6.46
	5	6	184	69.5	368	136.5	..	....	7.12
	6	0	199	75.0	398	146.5	..	....	7.67
4 Feet	3	6	156	62.0	312	120.5	46	50¼	5.94
	4	0	180	70.5	360	137.0	..	....	6.85
	4	6	200	77.5	400	151.0	..	....	7.60
	5	0	225	86.0	450	168.0	..	....	8.46
	5	6	249	94.5	498	184.5	..	....	9.37
	6	0	270	101.5	540	199.0	..	....	10.10
	6	6	294	110.0	588	215.5	..	....	11.01
	7	0	319	118.5	638	233.0	..	....	11.88
5 Feet	4	0	227	89.0	454	172.5	58	63¼	8.50
	4	6	253	98.0	506	190.5	..	....	9.45
	5	0	284	108.5	568	212.0	..	....	10.51
	5	6	314	119.0	628	233.0	..	....	11.68
	6	0	340	128.0	680	251.0	..	....	12.56
	6	6	371	138.5	742	272.0	..	....	13.65
	7	0	402	149.5	804	293.5	..	....	14.75
	7	6	427	158.0	854	310.5	..	....	15.75
	8	0	458	168.5	916	331.5	..	....	16.81
6 Feet	4	6	305	118.0	610	230.0	70	76¼	11.29
	5	0	342	131.0	684	255.5	..	....	12.60
	5	6	380	144.0	760	282.0	..	....	13.89
	6	0	411	155.0	822	303.5	..	....	14.96
	6	6	448	167.5	896	329.0	..	....	16.32
	7	0	485	180.5	970	354.5	..	....	17.64
	7	6	516	191.0	1032	376.0	..	....	18.75
	8	0	553	204.0	1106	401.5	..	....	20.07
	8	6	584	214.5	1168	422.5	..	....	21.18
	9	0	621	227.5	1242	448.0	..	....	22.50

Special Pipe Coil Heaters. There are several special forms of pipe coil heaters manufactured of which a typical example, designed to overcome certain difficulties encountered in the use of the standard type of heaters, is known as the *Positivflo* (Fig. 13). This heater is manufactured by the *Green Fuel Economizer Co.* In this type the horizontal tubes are attached at each end to a cast-iron box-header. One of these boxes is divided by a partition, as shown, so that the steam must flow in one direction through one half the tubes and in the opposite direction through the other half, the supply and return being connected to the same header. The principal advantage of this type is the ease with which leaky tubes in any section may be renewed.

Cast-Iron Indirect Heaters. The use of special cast-iron sections for indirect heaters is often practised, and Fig. 17 shows a cast-iron heating unit or section named *Vento,* manufactured by the *American Radiator Co.,* which is quite widely used in this class of work. A *stack* made up of several sections has a smaller number of joints compared with a pipe coil section of equal heating surface. The deterioration of the cast-iron sectional type of heater is practically nothing except for the right- and left-hand hexagonal nipples connecting the units which go to make up a stack.

Dimensions and Sizes of Vento Cast-Iron Heaters. There are three standard *lengths of*

sections available, each made in two different widths, *regular* and *narrow*. The following table gives the heating surface of these units. In addition to the three lengths given here in Table 6, there are also two others, one 30″ and the other 72″ long.

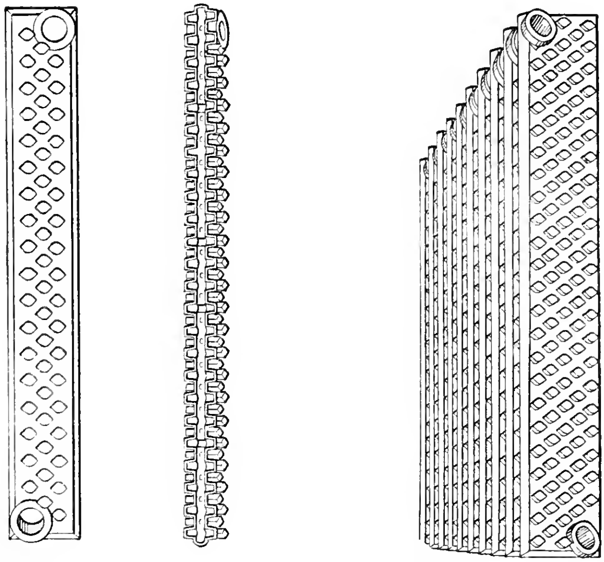


FIG. 17. VENTO HEATER SECTIONS.

TABLE 6

Length of Unit	Heating Surface, Sq. Ft.	
	Regular	Narrow
40″	10.75	7.50
50″	13.50	9.50
60″	16.00	11.00

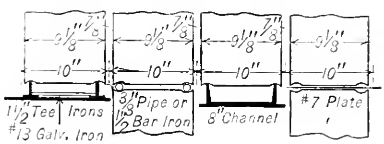


FIG. 18. TEE IRON SUPPORTS, VENTO HEATERS.
Light T-iron or Channel iron is used to support the Vento heaters as shown. This permits a slight movement of the several sections of the heater due to expansion.

The nipples, connecting the units, are made of such lengths that the spacing, center to center of units, may be $4\frac{5}{8}$″, 5″ or $5\frac{3}{8}$″, as may be desired.

The *clear or net free area* between the units, when assembled, is given by the following table:

TABLE 6-*a*

Length of Unit	Free Area Sq. Ft.		
	Centers		
	4⅝″	5″	5⅜″
40″	0.52	0.62	0.72
50″	.65	.77	.91
60″	.78	.92	1.08

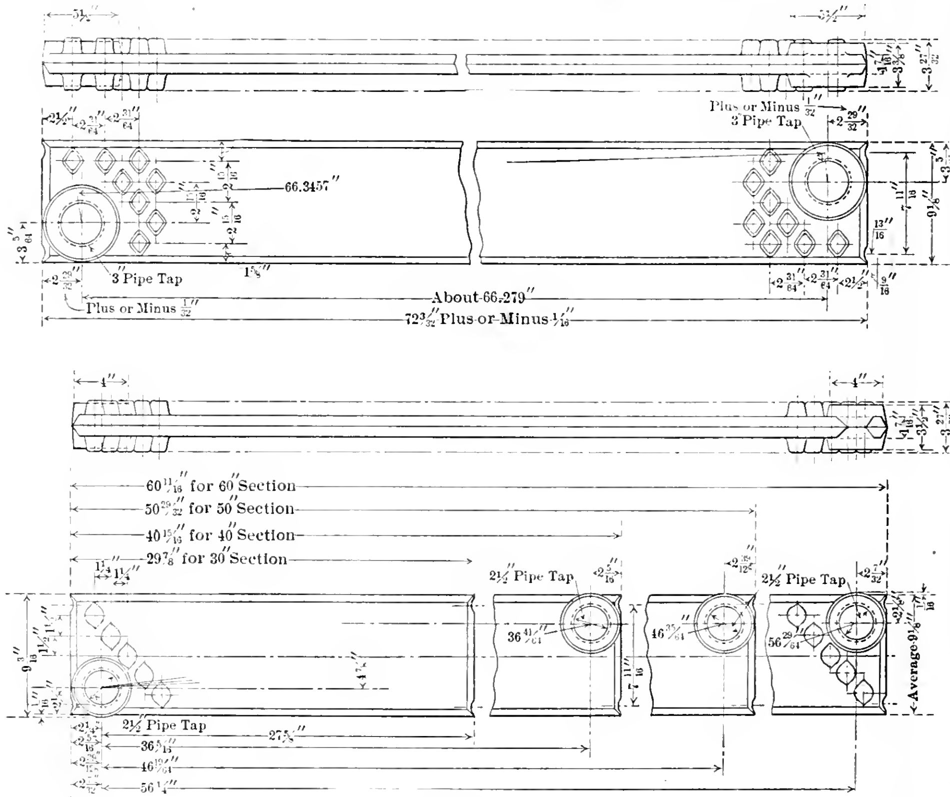


Fig. 19.

Note.—The sections of the Vento heater bear the trade names of 30, 40, 50, 60 and 72-in. sections, which are merely general designations and do not stand for the exact measurements of length.

TABLE 7

RATIO OF HEATING SURFACE TO FREE AREA. REGULAR VENTO SECTIONS, 5″ CENTERS

Length of Unit	$f = \dfrac{\text{Sq. Ft. H. S.}}{\text{Free Area}}$
40″	17.34
50″	17.53
60″	17.39

The method of assembling a "Vento" stack is clearly shown by Fig. 20, the loops being placed on 5-in. centers for the regular pattern. Fig. 21 shows, in plan, the assembly of a "Vento" heater made up of five stacks. It will be observed that the stacks are staggered. The

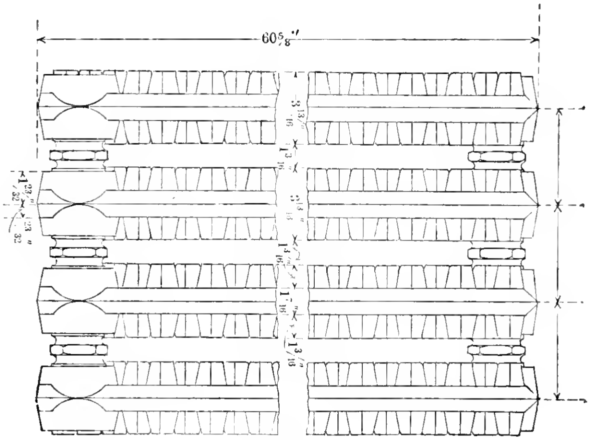


FIG. 20.

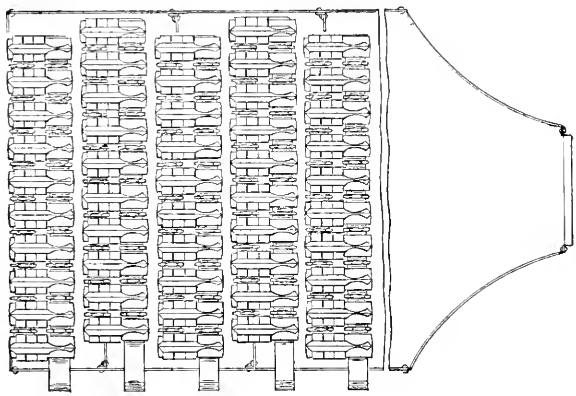


FIG. 21.

width of face, in inches, for a standard or regular section "Vento" heater is obtained by multiplying the number of sections or loops per stack by 5 and adding $2\frac{1}{2}$ in. for staggering. Thus the width of a heater having 10 regular sections or loops per stack is $(10 \times 5) + 2\frac{1}{2} = 52\frac{1}{2}$ in.

When the air is recirculated, as in factory heating, figure 25 per cent more surface for a given size of supply pipe.

When fresh air at 20 degrees below zero and colder is used, figure 15 per cent less surface for a given size of supply pipe.

TABLE 8

SIZES OF SUPPLY AND RETURN PIPES, VENTO HEATERS

Based on 1 to 5 pounds Steam Pressure. Runs not to exceed 100 feet.

	For Warming Fresh Air, Square Feet of Surface		Size of Supply	Size of Gravity Return	Size of Vacuum Return
	Tempering Coil	Re-Heater			
Connections to stacks	170	220	2 in.	1¼ in.	¾ in.
	270	360	2½	1½	1
	400	530	3	2	1¼
	Tempering Coils Only	Heating Work			
Mains supplying heater	550	730	3½ in.	2 in.	1¼ in.
	720	950	4	2½	1½
	1,125	1,500	5	3	2
	1,625	2,200	6	3½	2½
	2,425	3,250	7	4	3
	3,125	4,200	8	4	3½
	4,000	5,200	9	5	3½
	5,000	6,600	10	5	4
	7,000	9,500	12	6	5
	10,000	13,300	14	7	5

TABLE 9

FOR REDUCED STEAM PRESSURE OF 30 POUNDS

Square Feet of Surface	Size of Steam Main. Inches	Size of Return. Inches
450	2	1¼
650	2½	1½
1,000	3	2
1,350	3½	2
1,725	4	2½
2,700	5	3
4,000	6	3½

Author's Note.—It is recommended that the steam main, in every case, be designed according to some predetermined pressure drop. Ordinarily the amount of heating surface as called for by the above tables, for various pipe sizes, is exceeded in practice by 25 to 50 percent. A pressure drop of ½ to 1 lb. per 100 feet is frequently employed in this connection.

Piping Connections and Arrangements for Vento Heaters. The following charts, Figs. 22 to 25, are recommendations by the manufacturer, based on experience, for steam supply, return, and air-removal connections to Vento heaters. The arrangement of the stacks may be modified to suit local conditions.

HOT BLAST HEATER PERFORMANCES

Heat Transmission. The heat transmission of indirect pipe coil heaters has been found to depend upon the velocity of the air over the heater surface as well as the temperature difference between the surface and the air in contact with it. A film of air apparently exists next the heating surface which acts as an insulator and tends to reduce the heat emission. By rapidly circulating the air over the surface the insulating film is reduced in thickness and the heat emission increased. The heat transmission of pipe coil heaters is quite accurately given by a formula, proposed by *W. H. Carrier* deduced from the experiments of the *Buffalo Forge Co.*, and tests by the *American Radiator Co.*

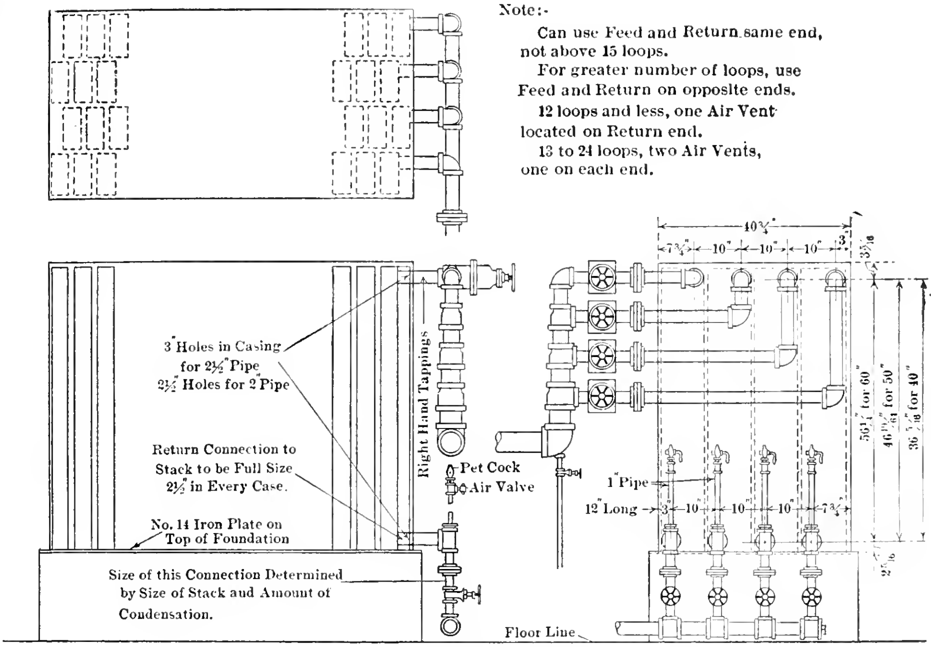


FIG. 22.

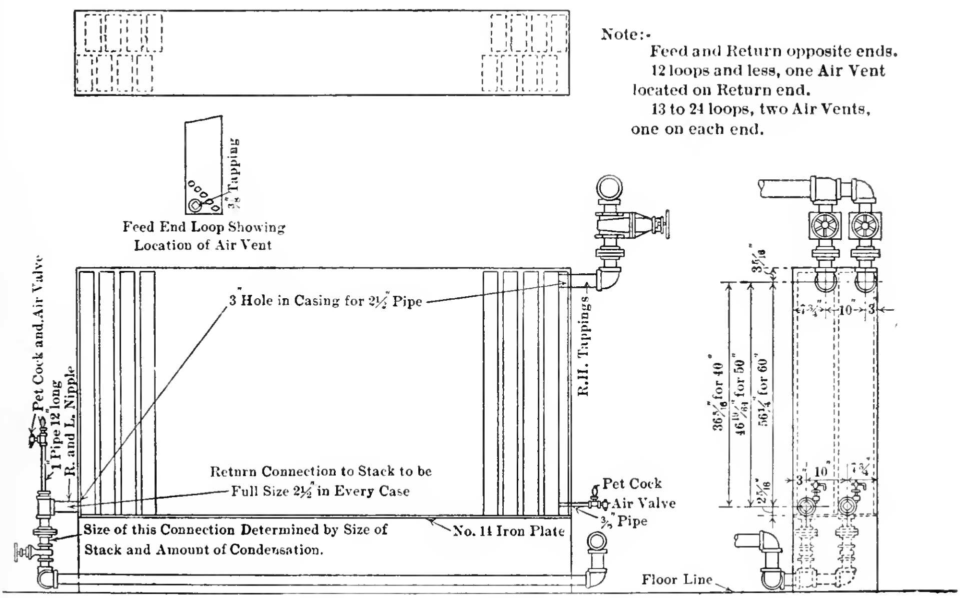


FIG. 23. GRAVITY SYSTEM CONNECTIONS.

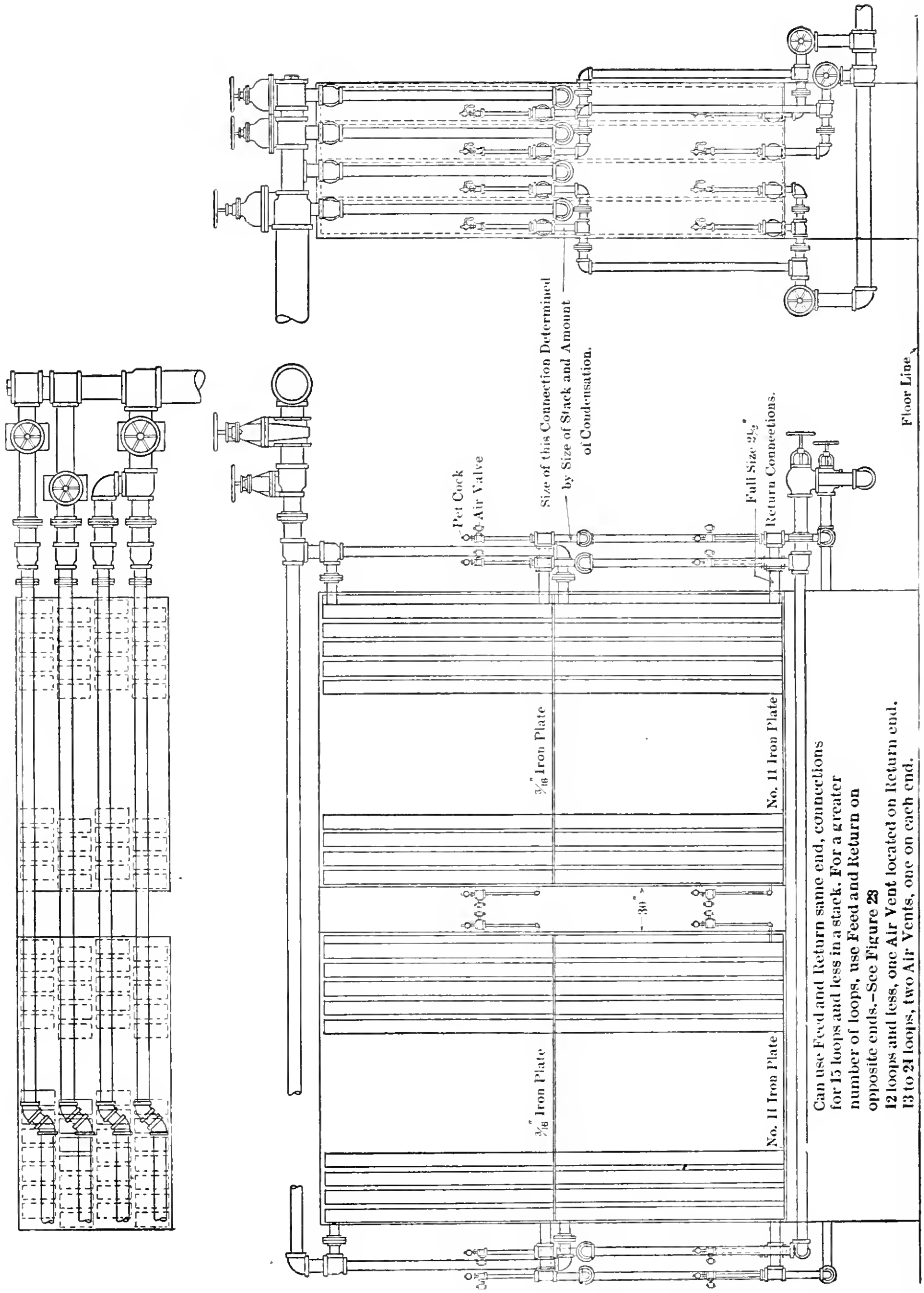


FIG. 24. GRAVITY SYSTEM CONNECTIONS.

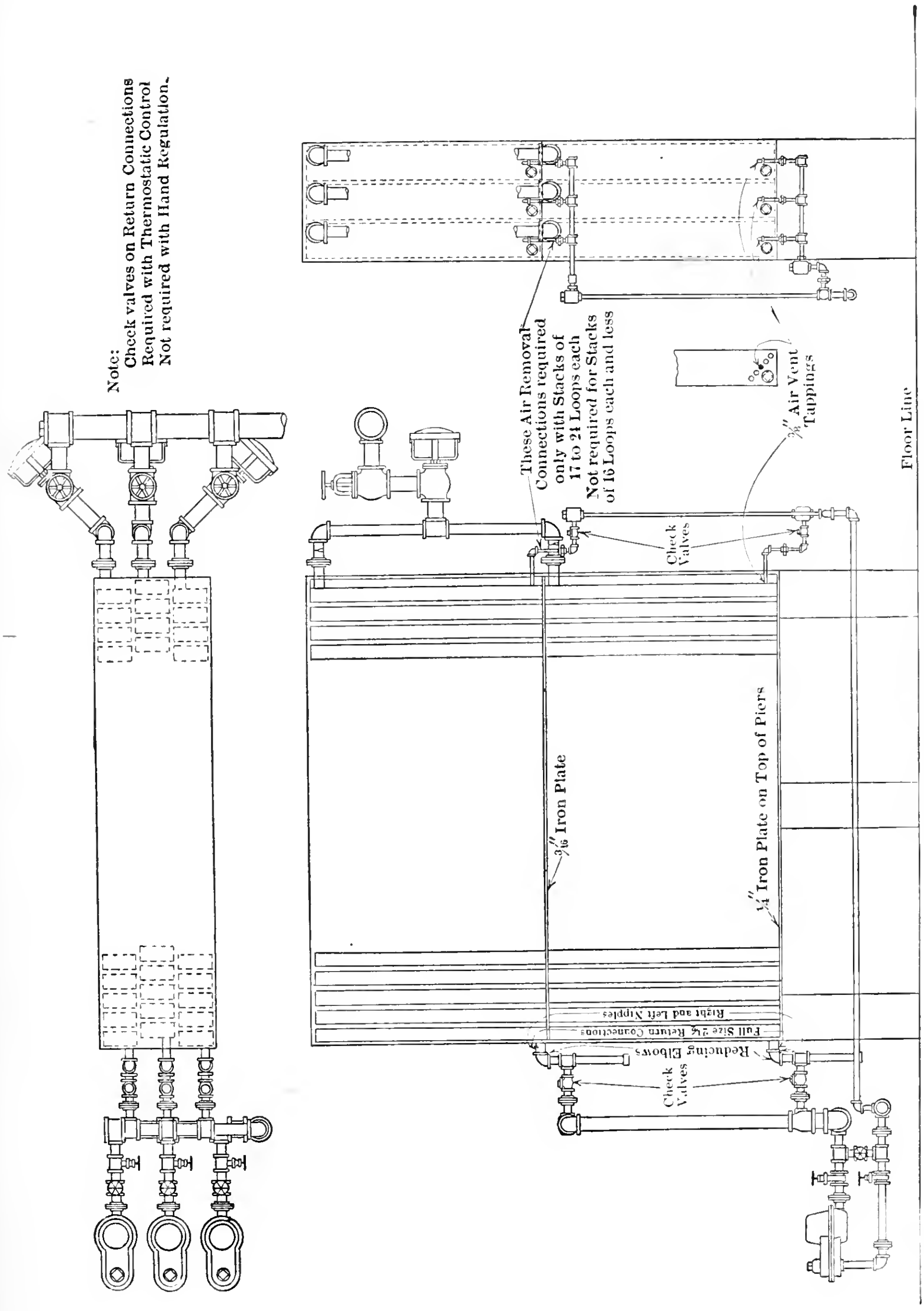


FIG. 25. VACUUM SYSTEM CONNECTIONS.

$$K = \frac{1}{0.0447 + \dfrac{50.66}{V}} \text{ for pipe coil heaters.}$$

$$K = \frac{1}{0.047 + \dfrac{61.00}{V}} \text{ for cast iron "Vento" sections, in which}$$

K = the heat transmission per sq. ft. per hour per degree difference in temperature between the air and steam.

V = velocity of the air over the heater surface in ft. per minute. This is the velocity through the *clear area* of heater. Values of "K" for various air velocities are given by curves, Fig. 26, for both types of heaters.

Mean Air Temperature.

Let t_1 = temperature of the entering air (initial temperature).
t_2 = temperature of the leaving air (final temperature).
t_m = mean temperature of the air.
t_s = temperature of the steam in coils.
$t_s - t_m$ = mean temperature difference between steam and air.

$$= \frac{(t_s - t_1) - (t_s - t_2)}{\log_e \left(\dfrac{t_s - t_1}{t_s - t_2}\right)} = \frac{(t_2 - t_1)}{2.3025 \log \left(\dfrac{t_s - t_1}{t_s - t_2}\right)}$$

Area of Heating Surface and Temperature Rise.

S = area of heating surface, sq. ft.
A = Free area through heater, sq. ft.
V = Velocity of air through free area measured at 70° F.
Q = Flow in cu. ft. per min. measured at 70° F.
= $A\ V$.
K = B.t.u. transmitted from the steam to air per sq. ft. per hour per degree difference in temperature (Fig. 26).
d = density of air at 70° F = 0.075.
$60 \times .075 \times A \times V$ = weight of air flowing per hour.
Then $K \times S \times (t_s - t_m) = 60 \times 0.075 \times A \times V \times 0.24\ (t_2 - t_1)$.

Substituting the value of $(t_s - t_m)$ from above we have $K \times S = 2.5 \times Q \times \log \left(\dfrac{t_s - t_1}{t_s - t_2}\right)$ (1)

Substituting the value of $K = \dfrac{1}{0.0447 + \dfrac{50.66}{V}}$ for pipe coil heaters

we have
$$S = (0.1118\ Q + 127\ A) \log \left(\frac{t_s - t_1}{t_s - t_2}\right) \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad (2)$$

for "Vento" sections
$$S = (0.2175\, Q + 152.5A) \log \left(\frac{t_s - t_1}{t_s - t_2}\right) \quad . \ . \ . \ . \ . \ . \ . \ . \quad (2\text{-}a)$$

If S_1 = Heating surface for one section = 4 rows pipe, usually, for a pipe-coil heater.

$f = \dfrac{S_1}{A}$ ratio of heating surface per section to the free area. This ratio varies somewhat with the different types of pipe coil heaters;

= 12.335 for standard return bend heaters, 1″ pipes placed on $2\frac{5}{8}$″ centers (*Buffalo Forge Co.*).

$= 17.45$ for " Vento " sections, 5" centers.

$S = N \times S_1$, total heating surface. N = number of sections deep.

$$f = \frac{S}{AN} \qquad S = f\,A\,N = A(0.1118V + 127)\log\left(\frac{t_s - t_1}{t_s - t_2}\right)$$

$$\text{or}\ \log\left(\frac{t_s - t_1}{t_s - t_2}\right) = \frac{12.34N}{0.1118\,V + 127}\ \text{for pipe coil heaters.} \qquad (3)$$

$$\text{Similarly for " Vento " sections,}\ \log\left(\frac{t_s - t_1}{t_s - t_2}\right) = \frac{17.45\,N}{0.1175V + 152.5} \qquad (4)$$

From equations (3) and (4) the temperature rise $(t_2 - t_1)$ for any condition may be calculated.

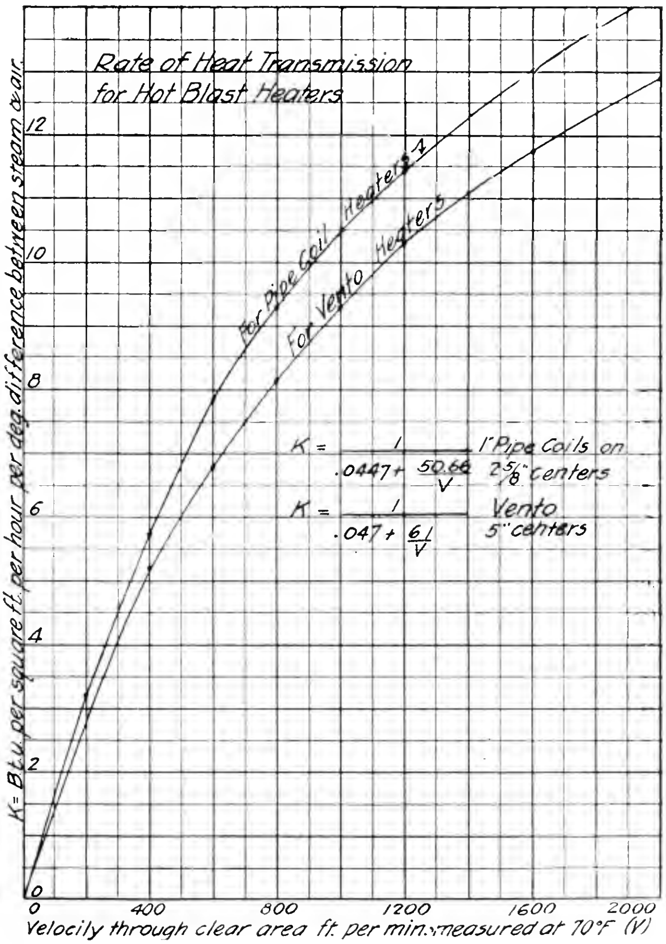


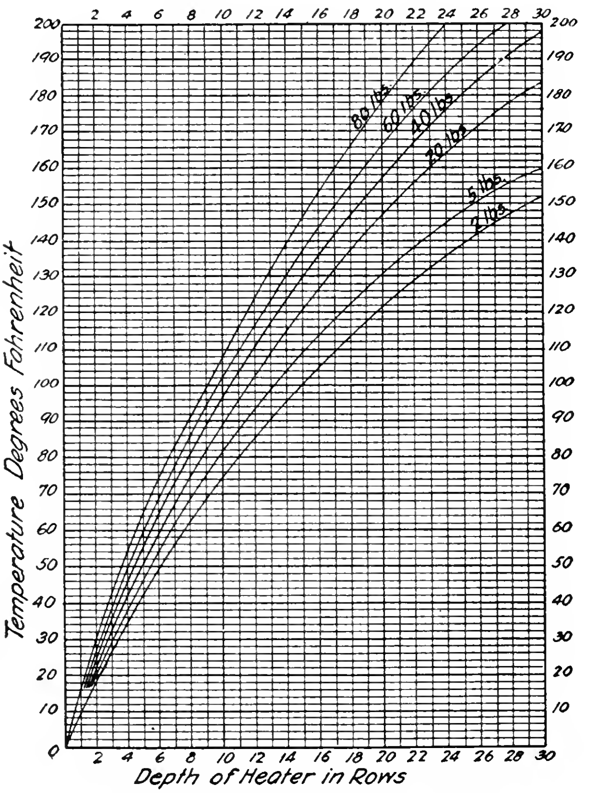
Fig. 26.

These formulas are general and may be applied to any type of hot blast heater when f is known for the particular heater.

Either equation (1) or (2) may be used in solving for the heating surface required for a

given temperature rise. Equation (3) may be used, when the relation between the heating surface and free area per section is known, to determine the temperature rise for a given number of rows of pipe.

Example. Required the amount of heating surface in a pipe coil heater, to raise the temperature of 37,037 lb. of air per hour from 0° to 155° F. Steam pressure 5 lb. gage. Velocity of air through clear area referred to 70° F. to be 1200 ft. per min. Volume of air measured at 70° F. is $\frac{37{,}037}{60 \times .075} = 8230$



Actual Temperature Increment at Velocity of 1200 Feet per minute. (B.F. Sturtevant Co.)

FIG. 27.

cu. ft. per min. $A = \frac{8230}{1200} = 6.86$ sq. ft. free area. $S = (0.1118 \times 8230 + 127 \times 6.86) \times \log \left(\frac{227 - 0}{227 - 155}\right) = 903$ sq. ft. of heating surface.

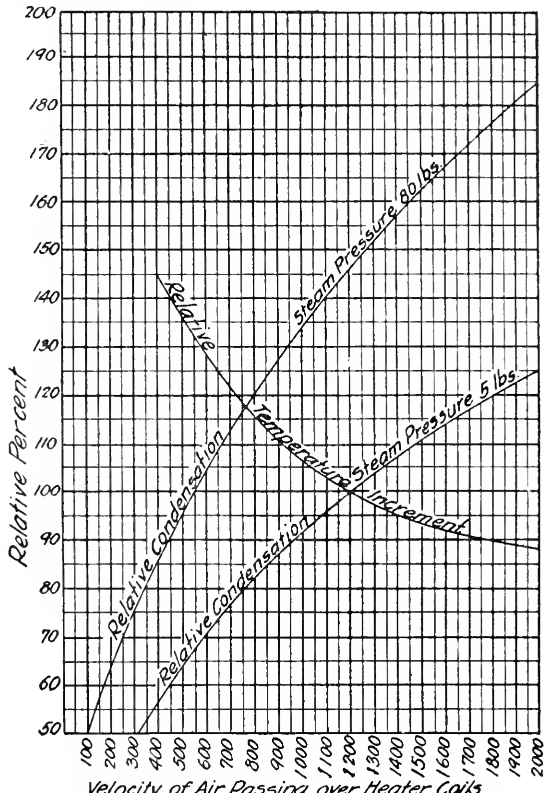
Referring to Table 2 (*Buffalo Forge Co.*) the nearest size section giving 6.8 sq. ft. for the free area is section number 44, containing 83.7 sq. ft. of heating surface. Therefore $\frac{903}{83.7}$ or 10 sections are required.

Example. Required the temperature rise above 0° for a six section pipe coil heater (24 rows of

pipe) having a ratio of $f = \frac{S_1}{A} = 12.33$ (*Buffalo Forge Co.*). Steam pressure 5 lb. gage. Air velocity 1200 ft. per min. referred to 70° F. Substituting the numerical values in equation (3) we have $\log\left(\frac{227-0}{227-t_2}\right)$

$$= \frac{12.335 \times 6}{0.1118 \times 1200 + 127} = 0.2832. \text{ Then } \left(\frac{227-0}{227-t_2}\right) = 1.92 \text{ or } t_2 = 108° \text{ F. final temperature.}$$

Compare with Mfg. No. 2, Table 10.



Effect of Velocity on the Relative Temperature Increment and Condensation. (B. F. Sturtevant Co.)

FIG. 28.

Required the temperature of a six section *B. F. Sturtevant Co.* heater in which $f = \frac{S_1}{A} = 19.7$ for the same conditions of operation.

$$\log\frac{227-0}{227-t_2} = \frac{19.7 \times 6}{0.1118 \times 1200 + 127} = 0.450, \text{ and } t_2 = 146°. \text{ Compare with diagram Fig. 27.}$$

Values of the final air temperature (t_2) obtained with various steam pressures for an air velocity of 1200 ft. per minute through the clear area of heater are given in Table 10. These data were obtained from the curves and published tables of the several manufacturers indicated and

and are based on an initial temperature $t_1 = 0$. In applying the published data of final temperatures obtained the method used in measuring or stating the velocity should be noted. In some cases the velocity refers to the initial velocity of the entering air, through the clear area, in others the average velocity, and to the velocity that would exist if the same weight of air passing through was at a temperature of 70°. Data on "Vento" heaters are given in Tables 11 and 11*a*.

TABLE 10

TEMPERATURE RISE OF AIR PASSING THROUGH HEATERS—INITIAL TEMPERATURE OF AIR 0° F.

Steam Pressure, Pounds Gage	Mfgr.	Velocity 1200 Ft. per Min. No Sections or Stacks						
		2	3	4	5	6	7	8
2	1	63	85	103	118	132	142	152
	3	63	85	105	121	135	147	...
	4	60	78	96	113	128	143	157
4	1	65	88	107	123	137	147	158
5	2	43	61	79	92	108	119	130
	3	69	93	113	130	145	155	...
	4	64	84	102	120	135	150	162
	5	58	81	100	115	129	140	150
20	1	75	101	122	140	156	168	180
	2	52	73	92	109	124	138	150
	3	75	102	126	147	163	177	...
	4	70	91	113	130	147	162	175
30	5	73	101	124	143	159	173	184
60	1	87	120	145	166	185	199	213
	2	60	85	107	127	145	161	175
	3	86	116	143	166	185	200	...
	4	82	109	134	153	170	185	197

1. *American Blower Co.* Velocity is average velocity through coils.
2. *Buffalo Forge Co.* Velocity measured at 70° F. 1″ pipes on $2\frac{5}{8}''$ centers.
3. *B. F. Sturtevant Co.* Velocity measured at 70° F. 1″ pipes, $2\frac{1}{8}''$ on centers.
4. "Positivflo"—*Green Fuel Economizer Co.* Velocity is initial velocity of entering air.
5. "Vento"—*American Radiator Co.* Reg. Section 5″ centers. Velocity measured at 70° F.

SELECTION OF HEATER

General Conditions. In selecting the size of a heater for any particular service the choice is based on the final temperature desired and the "free area" required for a certain allowable velocity. That is, for a specified initial and final temperature desired, and a certain number of sections, a final temperature results when the velocity has been fixed in advance. Good practice limits the velocity to the values given by the following tables. High velocities are objectionable in public building work on account of the noise resulting. The resistance through the heater increases in proportion to the square of the velocity which adds to the power required to move the air as the velocity is increased, as will be noted later.

Free Area Required.

Let M = total weight of air flowing through heater per hour.

V_1 = velocity of entering air ft. per min. through free area.

V_2 = velocity of leaving air ft. per min. through free area.

$V_a = \dfrac{V_1 + V_2}{2}$ = average velocity air ft. per min. through free area.

TABLE 11

FINAL TEMPERATURES AND CONDENSATIONS—VENTO HEATERS

Regular Section—Standard spacing, 5″ Centers of Loops—Steam 227°, 5 lbs. Gage. Air entering above zero

Number of Stacks Deep	Temperature of Entering Air	Velocity Through Heater in Feet per Minute—Measured at 70°																			
		100		120		200		600		800		1000		1200		1400		1600		1800	
		Final Temp. Air Leaving Heater	Cond. Lb. Per Sq. Ft. Per Hour	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.
	20	87	.43	83	.48	75	.70	58	1.46	54	1.75	51	1.99	49	2.23	47	2.42	45	2.56	43	2.65
	30	93	.40	89	.45	81	.65	66	1.39	62	1.64	60	1.92	58	2.17	56	2.33	54	2.46	52	2.54
1	40	100	.38	96	.43	88	.62	74	1.31	70	1.54	68	1.80	66	2.00	64	2.16	62	2.26	61	2.42
	60	112	.33	109	.38	102	.54	90	1.15	86	1.34	84	1.54	82	1.69	81	1.89	80	2.05	79	2.19
	70	118	.31	115	.35	109	.50	97	1.04	94	1.23	92	1.41	90	1.54	89	1.71	88	1.85	87	1.96
	20	129	.35	124	.40	112	.59	87	1.29	81	1.57	76	1.80	72	2.00	69	2.20	66	2.36	64	2.54
	30	133	.33	129	.38	117	.56	93	1.21	87	1.46	83	1.70	79	1.89	76	2.06	73	2.21	71	2.37
2	40	137	.31	133	.36	121	.52	100	1.15	94	1.39	90	1.60	86	1.77	83	1.93	81	2.10	79	2.27
	60	145	.27	142	.32	131	.46	112	1.00	107	1.21	103	1.38	100	1.54	98	1.71	96	1.85	94	1.96
	70	149	.25	146	.29	136	.42	118	.92	114	1.13	110	1.28	107	1.42	105	1.57	103	1.69	101	1.79
	20	157	.29	152	.34	139	.51	110	1.15	103	1.42	97	1.65	92	1.85	88	2.06	85	2.22	82	2.38
	30	160	.28	155	.32	142	.48	115	1.09	108	1.33	103	1.56	98	1.75	94	1.91	91	2.08	88	2.23
3	40	163	.26	158	.30	146	.45	121	1.04	114	1.26	109	1.47	104	1.64	100	1.79	97	1.95	94	2.08
	60	168	.23	164	.27	153	.40	131	.91	124	1.09	120	1.28	116	1.44	113	1.58	110	1.71	108	1.85
	70	170	.21	167	.25	156	.37	136	.85	130	1.03	126	1.20	122	1.34	119	1.46	116	1.57	114	1.69
	20	...	...	...	...	...	...	130	1.06	122	1.31	115	1.52	110	1.73	105	1.91	101	2.08	97	2.22
	30	...	...	...	...	...	...	134	1.00	126	1.23	120	1.44	115	1.63	110	1.80	106	1.95	102	2.08
4	40	...	...	...	...	...	...	138	.94	130	1.15	124	1.35	119	1.52	115	1.68	111	1.82	108	1.96
	60	...	...	...	...	...	...	146	.83	139	1.01	134	1.19	129	1.33	125	1.46	122	1.59	119	1.70
	70	...	...	...	...	...	...	150	.77	143	.94	138	1.09	134	1.23	131	1.37	128	1.49	125	1.59
	20	...	...	...	...	...	...	144	.95	136	1.19	130	1.41	124	1.60	119	1.78	114	1.93	110	2.08
	30	...	...	...	...	...	...	148	.91	140	1.13	134	1.33	128	1.51	123	1.67	118	1.80	115	1.96
5	40	...	...	...	...	...	...	151	.85	144	1.07	138	1.26	132	1.42	127	1.56	123	1.70	120	1.85
	60	...	...	...	...	...	...	158	.75	151	.93	145	1.09	140	1.23	136	1.36	133	1.50	130	1.62
	70	...	...	...	...	...	...	162	.71	155	.87	149	1.01	144	1.14	141	1.27	138	1.40	135	1.50
	20	...	...	...	...	...	...	156	.87	148	1.10	142	1.30	136	1.49	130	1.65	126	1.81	122	1.96
	30	...	...	...	...	...	...	159	.83	151	1.04	145	1.23	139	1.40	134	1.56	130	1.71	126	1.85
6	40	...	...	...	...	...	...	162	.78	151	.97	148	1.15	143	1.32	138	1.47	134	1.60	131	1.75
	60	...	...	...	...	...	...	167	.69	160	.85	155	1.02	150	1.15	146	1.29	142	1.40	139	1.52
	70	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	20	...	...	...	...	...	...	167	.81	159	1.02	152	1.21	146	1.39	141	1.55	136	1.70	132	1.85
	30	...	...	...	...	...	...	169	.76	161	.96	155	1.15	149	1.31	144	1.46	139	1.60	135	1.73
7	40	...	...	...	...	...	...	171	.72	164	.91	158	1.08	153	1.24	148	1.39	143	1.51	139	1.63
	60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	70	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	20	...	...	...	...	...	...	175	.75	167	.94	161	1.13	155	1.30	150	1.46	145	1.60	141	1.74
	30	...	...	...	...	...	...	177	.71	169	.89	163	1.07	158	1.23	153	1.38	148	1.51	144	1.64
8	40	...	...	...	...	...	...	179	.67	171	.84	165	1.00	160	1.15	155	1.29	151	1.42	147	1.54
	60	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	70	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...

TABLE 11*a*

FINAL TEMPERATURES AND CONDENSATIONS—VENTO HEATERS

Regular Section—Standard spacing, 5″ Centers of Loops—Steam 227°, 5 lbs. Gage. Air entering below zero.

Number of Stacks Deep	Temperature of Entering Air	Velocity Through Heater in Feet per Minute—Measured at 70°																					
		100		120		200		600		800		1000		1200		1400		1600		1800		2000	
		Final Temp. Air Leaving Heater	Cond. Lbs. Per Sq. Ft. Per Hour	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.	F.T.	C.
	−20	61	.51	61	.62	48	.87	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
1......	−10	70	.51	67	.59	55	.83	34	1.69	...	...	...	...	...	...	...	...	...	...	...	...	...	...
	0	76	.49	73	.56	62	.79	43	1.65	38	1.95	35	2.24	32	2.46	...	...	...	...	...	...	...	...
	−20	113	.43	108	.49	93	.72	63	1.60	55	1.92	49	2.22	44	2.46	40	2.69	37	2.92	34	3.12	31	3.27
2......	−10	117	.41	112	.47	97	.69	69	1.52	62	1.85	56	2.12	51	2.35	47	2.56	44	2.77	41	2.94	38	3.08
	0	121	.39	116	.45	102	.65	75	1.44	68	1.74	62	1.99	58	2.23	54	2.42	51	2.62	48	2.77	46	2.95
	−20	146	.35	142	.42	125	.62	91	1.42	82	1.74	75	2.03	69	2.28	64	2.51	59	2.70	55	2.88	52	3.08
3......	−10	149	.34	145	.40	129	.59	96	1.36	87	1.66	80	1.92	75	2.18	70	2.39	66	2.60	62	2.77	58	2.90
	0	152	.33	148	.38	132	.56	101	1.30	93	1.59	86	1.84	81	2.08	76	2.27	72	2.46	68	2.62	65	2.78
	−20	...	...	...	...	...	...	114	1.29	103	1.58	96	1.86	90	2.12	84	2.34	78	2.51	74	2.71	70	2.88
4......	−10	...	...	...	...	...	...	117	1.22	108	1.51	101	1.78	95	2.02	89	2.22	84	2.41	80	2.60	76	2.76
	0	...	...	...	...	...	...	121	1.16	113	1.45	106	1.70	100	1.92	95	2.13	90	2.31	86	2.48	82	2.63
	−20	...	...	...	...	...	...	132	1.17	122	1.46	114	1.72	107	1.95	100	2.15	94	2.34	90	2.54	86	2.72
5......	−10	...	...	...	...	...	...	135	1.13	126	1.40	118	1.64	111	1.86	105	2.06	99	2.24	95	2.42	91	2.59
	0	...	...	...	...	...	...	138	1.06	129	1.32	122	1.56	115	1.77	109	1.96	104	2.14	100	2.31	96	2.46
	−20	...	...	...	...	...	...	146	1.06	137	1.34	129	1.59	121	1.81	115	2.02	110	2.22	105	2.40	100	2.56
6......	−10	...	...	...	...	...	...	149	1.02	140	1.28	132	1.52	125	1.73	119	1.93	114	2.12	109	2.29	104	2.44
	0	...	...	...	...	...	...	152	.97	143	1.22	135	1.44	129	1.65	123	1.84	118	2.02	113	2.17	109	2.33
	−20	...	...	...	...	...	...	159	.98	150	1.25	141	1.47	134	1.69	128	1.90	122	2.08	117	2.26	113	2.44
7......	−10	...	...	...	...	...	...	161	.94	152	1.19	144	1.41	137	1.62	131	1.81	126	1.99	121	2.16	117	2.33
	0	...	...	...	...	...	...	163	.90	154	1.13	147	1.35	140	1.54	135	1.73	130	1.90	125	2.06	121	2.22
	−20	...	...	...	...	...	...	168	.90	159	1.15	151	1.37	144	1.58	138	1.77	133	1.96	128	2.14	123	2.29
8......	−10	...	...	...	...	...	...	170	.87	161	1.10	153	1.31	147	1.51	141	1.69	136	1.87	131	2.04	126	2.18
	0	...	...	...	...	...	...	172	.83	164	1.05	156	1.25	150	1.44	144	1.62	139	1.78	134	1.93	129	2.07

TABLE 12

ALLOWABLE VELOCITIES OF AIR THROUGH VENTO HEATERS

Referred to a temperature of 70° F.

Number of Stacks Deep, Regular 5-Inch Centers	Public Building Work, Velocity in Feet per Minute	Factory Work, Velocity in Feet per Minute
4	1000 to 1500	1200 to 1600
5	1000 to 1300	1200 to 1600
6	1000 to 1200	1200 to 1600
7	900 to 1100	1200 to 1500
8	800 to 1000	1200 to 1400

TABLE 13

RECOMMENDED VELOCITIES OF AIR THROUGH PIPE COIL HEATERS

(*Buffalo Forge Co.*)

Referred to a temperature of 70° F.

Number Sections	Public Buildings, Feet per Minute	Industrial Plants, Feet per Minute
4	1140	1500
5	1020	1350
6	930	1230
7	860	1140
8	810	1070

V_{70} = velocity referred to a temp. of 70° F.
d_1 = density of entering air.
d_2 = density of leaving air.
0.075 = density at 70°.
A = free area required in sq. ft.

$$A = \frac{M}{60 d_1 V_1} \text{ when the rating is based on initial velocity.}$$

$$A = \frac{M}{60 \times 0.075 \times V_{70}} \text{ when the rating is referred to a temperature of 70° F.}$$

$$A = \frac{M}{60 \times \frac{d_1 + d_2}{2} \times V_a} \text{ when the rating is based on average velocity.}$$

The following table will serve as an appropriate guide in choosing the number of heater sections required for various types of service.

TABLE 14

NUMBER OF HEATER SECTIONS REQUIRED

Service	No. Sections	No. Rows 1″ Pipe
Public Buildings Fresh Air, Exhaust Steam	5	20
Industrial Buildings Fresh Air, 4 Lb. Gage	6	24
Industrial Buildings Fresh Air, 60 Lb. Gage	5	20
Industrial Buildings Fresh Air, Exhaust Steam	7	28
Industrial Buildings Recirculation Exhaust Steam	5	20
Tempering Coils Fresh Air, Exhaust Steam	2	8

When a tempering coil is employed deduct two sections from the number above specified for the heater.

Heater Condensation. The weight of steam, in lb. per hour, required by the heater may be calculated as follows:

Let W = weight of steam to be supplied per hour (dry and saturated).
h = total heat corresponding to the steam pressure at heater.
q = heat of liquid of condensate discharged.
r = latent heat.
R = temperature rise of air passing through heater $(t_2 - t_1)$.

Then $W = \dfrac{0.24\ R\ M}{h - q}$. It is customary to assume that the heat of the liquid discharged is the same as the heat of the liquid corresponding to the steam pressure. Then $h - q = r$.

An allowance of approximately 5 per cent should ordinarily be made for condensation in the supply main.

Example. (*Selection of Heater.*) Let it be required to select a pipe coil heater for the following conditions of operation of a factory installation. Steam at 5-lb. gage pressure to be supplied heater. All air to be recirculated, inside temperature to be maintained 65° F. Total heat loss of building H = 800,000 B.t.u. per hour.

The initial temperature of air entering heater is 65° F. A 5-section heater (20 rows of pipe) will be selected (see Table 14) with a velocity through the clear area of 1200 ft. per min. referred to a temperature of 70° F. Referring to the 5-lb. curve (Fig. 27), *B. F. Sturtevant Co.* pipe coil heaters, we find that it would require 8 rows (nearest even number) to heat the air from 0 to 65° F. The total number of rows to be actually used is 20. And 8 + 20 or 28 rows would raise the temperature from 0 to 155° F. Therefore a 5 section heater or 20 rows of pipe will raise the temperature from 65° F to 155 F° or 90° F.

The weight of air to be circulated per hour is:

$$M = \frac{H}{0.24\ (t_h - t)} = \frac{800{,}000}{0.24(155 - 65)} = 37{,}037 \text{ lbs.}$$

The "free area" required is: $A = \dfrac{M}{60 \times 0.075 \times V_{70}} = \dfrac{37{,}037}{60 \times 0.075 \times 1200} = 6.86$ sq. ft.

Referring to Table 5 we find that a $4' - 0'' \times 4' - 0''$ section is required.

If the apparatus is to be of the "draw through" type the fan must be able to take care of 37,037/[60 × 0.065(density at 155°)]or 9500 cu. ft. per minute.

THE FLOW OF AIR UNDER LOW PRESSURES AND ITS MEASUREMENT

The laws governing the flow of all fluids are based on the assumption that the density remains constant throughout the flow. In considering the flow of a gas such as air, however, the laws referred to do not strictly hold. The velocity in an air duct of uniform size varies due to a loss or decrease in pressure which causes an increase in volume and a consequent increase in the velocity.

The flow of air due to a large difference in pressure is most accurately stated by thermodynamic formulas for air discharge under conditions of adiabatic flow.

The flow of air in heating and ventilating ducts takes place under *very small pressure differences* usually less than a pressure equivalent of 2″ of water. Under this condition the change in density is very slight and for all practical purposes of calculation and design the density may be assumed constant and *the laws governing the flow of fluids applied.*

The flow of air in a duct or pipe is under the influence of three distinct pressures namely, the *velocity, static,* and *dynamic* or *total* pressures.

The *velocity head or pressure* (*V. P.*) is defined as that pressure which is required to create

the velocity of flow. That is the pressure or head required to accelerate the mass from a state of rest to the final velocity attained.

The *static head or pressure* (*S. P.*), also termed the frictional or resistance pressure or maintained resistance, is that pressure required to overcome the resistance offered to the flow. This, in reality, is the pressure tending to burst the pipe as would be measured by the ordinary pressure gage.

The *dynamic head or pressure* (*T. P.*), also termed the total or impact pressure, is the sum of pressures required to overcome the resistance to flow and create the velocity of discharge. That is $T.\ P. = S.\ P. + V.\ P.$ or $V.\ P. = T.\ P. - S.\ P.$

The usual method of stating and measuring the small pressure above atmospheric existing

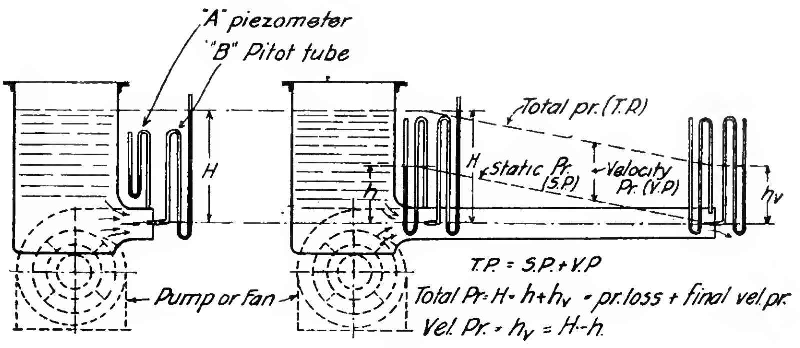


FIG. 29.

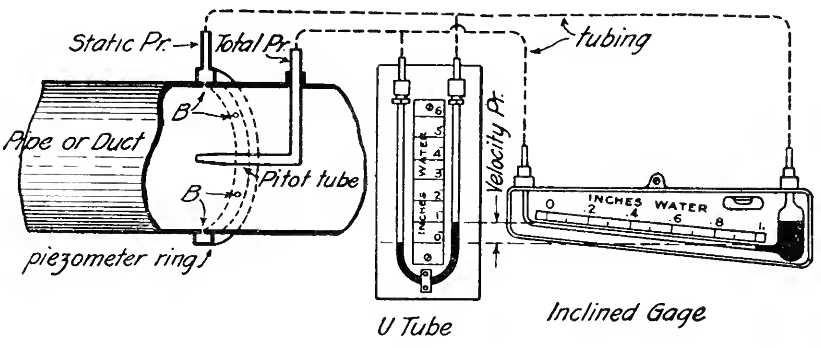


FIG. 30.

in heating or ventilating systems is made by noting the height of a column of water which the pressure will maintain in equilibrium or balance in a U-tube or manometer.

Piezometer. Referring to Fig. 29, tube A is inserted at right angles to the flow. The difference in level of the liquid in the manometer is a measure of the static pressure existing at this point in the pipe or duct. A tube of this description is termed a *Piezometer*. In order to obtain a correct piezometer reading the most accurate method is to employ a hollow ring, Fig. 30, connected to the interior of the pipe by six or eight small holes *B* (approximately .02″ diameter), and the manometer as shown.

Some form of differential or inclined tube gage filled with gasolene and graduated to read direct in hundredths of an inch of water is usually employed in place of the ordinary U-tube, which is not well adapted for reading small pressure differences.

Pitot Tube. A tube B (Fig. 29), bent at the end so that the opening faces the flow squarely, is termed a *Pitot tube.* The manometer, or differential gage connected to this tube measures the total or dynamic pressure existing at this point in the duct.

If the *Piezometer* and the *Pitot tube* be connected to opposite ends of the U-tube or differential gage, the difference in elevation of the liquid will now give the velocity head or pressure in inches of water.

As was previously shown under the flow of water the velocity of flow and the head or pressure required to produce the flow are related by the following formula:

$$v = \sqrt{2gh_x} \text{ or } h_x = \frac{v^2}{2g}$$

in which v = the velocity in ft. per sec. and h_x = the height in feet of the medium flowing, or velocity head or pressure.

As the actual measurement of head is made in inches of water, and not in feet of air column, it is necessary to know the relation between the two which follows.

Let K = density of water.
= 62.4 lb. per cu. ft. at 70° F.
d = density of air flowing corresponding to the temperature and pressure.
h_v = velocity head measured in inches of water equal to the difference between the total and static head.

$$\text{Then } 12dh_x = h_v k \text{ or } h_x = \frac{h_v k}{12d} = \frac{v^2}{2g}$$

$$\text{or} \qquad h_v = \frac{12dv^2}{2gk} \text{ or } v = \sqrt{\frac{2gh_v k}{12d}}$$

$$v = 18.27\sqrt{\frac{h_v}{d}} \qquad (1)$$

For dry air at 70° F. and 29.92 in. barometer d = 0.0749

$$\text{Then } v = 66.75\sqrt{h_v} \qquad (2)$$

The density for any temperature and pressure is readily determined from the relation.

$P\,V = M\,R\,T$ (Chapter II).
P = absolute pressure in lb. per sq. ft. = 144 × 14.7 or 2116.8 lb. at sea level.
V = volume in cu. ft.
M = weight of air in lb. = 1.
T = absolute temperature degs. F.
= $t + 460$.
R = 53.35.

$$\text{Then } d = \frac{1}{V} = \frac{P}{R\,T} = \frac{39.7}{T}. \quad \text{For 70° F. air } d = \frac{39.7}{70 + 460} = 0.0749.$$

The following diagram (Fig. 31) gives the relation between the velocity and velocity head or pressure for 70° F. air as calculated by the above equation.

Measurement of Air Flow with Pitot Tube. The *Piezometer* and *Pitot tube* are combined in one instrument, as shown by Fig. 32, which is the standard *American Blower Co.* tube used for fan testing. The outside or static tube is closed except for the small fine holes on the sides as shown; the small inside pipe being the *Pitot tube.* The inside and outside tubes terminate in the vertical tubes which are connected to the differential gage by means of rubber hose. It is essential that the hose connections be absolutely air tight for accurate results.

The velocity of air through a pipe or duct is not uniform over the cross section owing to the retarding action produced by the friction of air on the inside surface. The velocity is great-

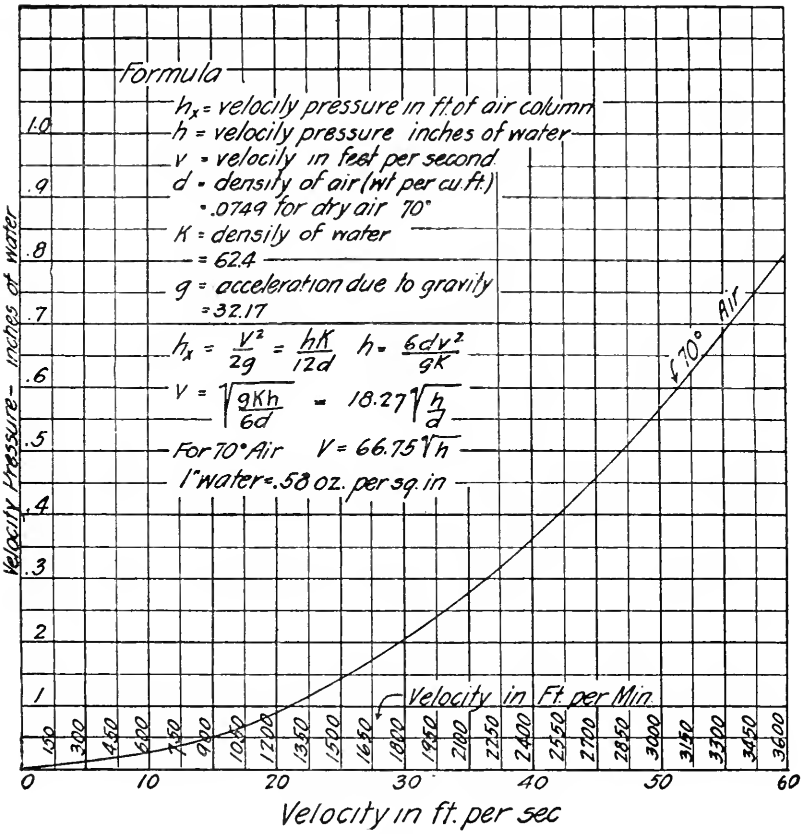


Fig. 31.

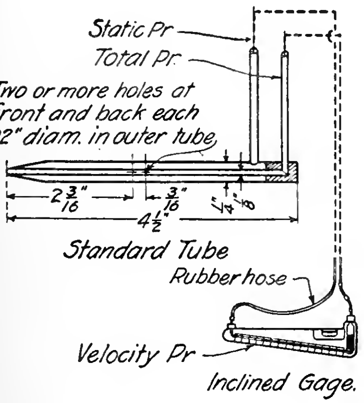


Fig. 32

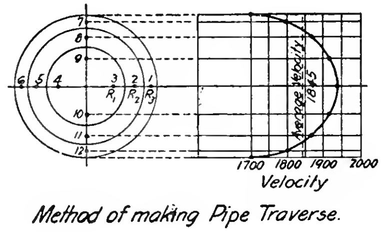


Fig. 33.

est at the center and least at the pipe surface. In order to obtain the actual flow, the following method is used: (Fig. 33.)

A round pipe is divided into at least three concentric zones of equal area *per foot in diameter* and four readings taken on a circle drawn through the center of area of each zone or ring. That is, readings should be taken across the horizontal and vertical axes of the pipe as shown by Fig. 33. The location of those points from the center is shown, together with the accompanying Table 15 from *Transactions A. S. H. and V. E., 1914*. This table gives the distance from the center of the pipe to the point of reading in per cent of the pipe diameter.

TABLE 15

PIPE TRAVERSE FOR PITOT TUBE READINGS. DISTANCE FROM CENTER OF PIPE TO POINT OF READING IN PER CENT OF PIPE DIAMETER

No. of Equal Areas in Traverse	No. of Readings	1st R_1	2d R_2	3d R_3	4th R_4	5th R_5	6th R_6	7th R_7	8th R_8
3	12	20.4	35.3	45.5	....	....	....	....	....
4	16	17.7	30.5	39.4	46.6	....	....	....	....
5	20	15.8	27.2	35.3	41.7	47.4	....	....	....
6	24	14.5	25.0	32.3	38.2	43.3	47.9	....	....
7	28	13.4	23.1	29.9	35.3	40.1	44.3	48.2	....
8	32	12.5	21.6	28.0	33.2	37.6	41.5	45.1	48.4

The corresponding velocity for each reading should be calculated from the formula $V = 18.27\sqrt{\frac{h_v}{d}}$ and the average of the sum of the velocities taken for the mean velocity. Thus the mean velocity for a pipe divided into three equal areas would be:

$$V_m = \frac{V_1 + V_2 + V_3 + \ldots\ldots + V_{12}}{12} \qquad (3)$$

Inasmuch as the velocity varies as the square root of the velocity pressure or head, accurate results cannot be obtained by taking the average of the pressure readings and taking the corresponding velocity as the average.

Replacing the velocities in equation (3) by their equivalents we have for any number of readings,

$$V_m = \frac{18.27}{d^{1/2}} \; \frac{(\sqrt{h_1} + \sqrt{h_2} + \sqrt{h_2} + \ldots\ldots + \sqrt{h_n})}{n\ (no.\ \text{of readings})} \qquad (4)$$

For approximate results, with circular pipe $V_m = 0.91$ to $0.93 \times$ velocity determined from the reading at the center of the pipe.

In making a traverse of a rectangular or square pipe or duct the cross sectional area may be divided into a number of small rectangles or squares of equal area and a reading taken at the center of each small rectangle or square.* The corresponding velocity for each reading is calculated and the average taken as for circular pipe.

The tube should, when possible, be located at least 10 diameters of the pipe from the fan outlet, from an elbow, or from a change in cross section of the pipe.

Let Q = volume of air flowing cu. ft. per sec.

V_m = mean velocity in ft. per sec.

A = Area pipe sq. ft.

Then $Q = A\ V_m$.

*Note.—For recommended practice in making a traverse of a square or rectangular pipe see the report of a special committee appointed by the *A. S. H. and V. E.*, January, 1913.

The Pitot tube is not applicable to measuring velocities below approximately 400 ft. per min. or 6.6 ft. per sec. If used with proper care and accuracy of reading the Pitot tube gives results with an error of less than 1½ per cent with velocity pressures ranging from 0.1 in. and upwards.

The Anemometer. For velocities below 1500 ft. per min. an anemometer (Fig. 34) may be used at the end of a pipe or at a register face. It is necessary frequently to calibrate this instrument in order to obtain anything like accurate results.

Calorimeter Method of Measuring Air Flow. If air flowing through a pipe be heated by

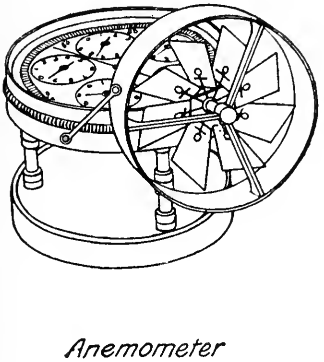


FIG. 34.

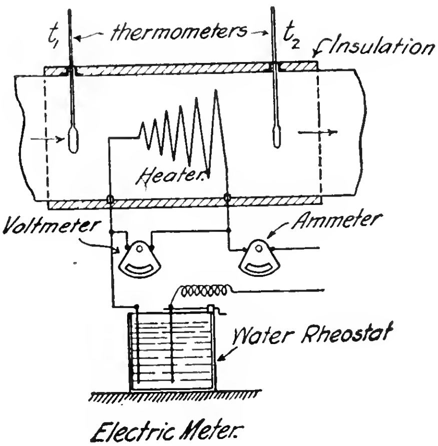


FIG. 35.

passing it over an electrical or steam heater, noting the temperature rise and the heat input per hour, the weight of air flowing is readily determined by the following relation: (See Fig. 35.)

Let H = heat input B.t.u. per hour.
t_1 = initial temperature of air.
t_2 = final temperature of air.
0.24 = specific heat of air, constant pressure.
M = weight of air flowing per hour.

$$H = 0.24\,M\,(t_2 - t_1) \text{ or } M = \frac{H}{0.24(t_2 - t_1)}.$$

If an electrical heater is used,

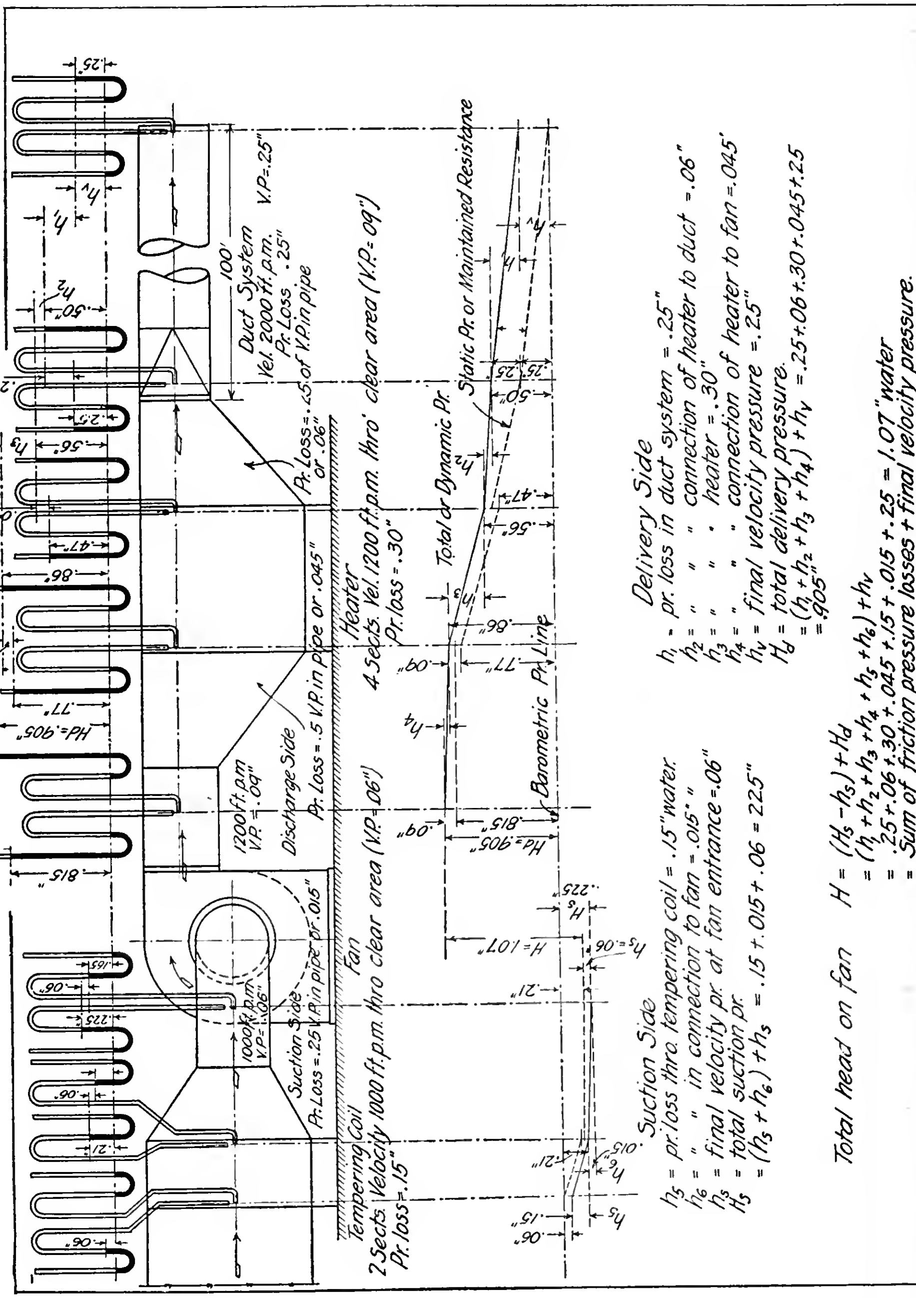
$$H = 3.415 \times \text{watts input}.$$

If a steam heater is used and the weight of steam condensed per hour, corrected for moisture, is W lb.

Then $H = W\,(h - q)$, in which h is the total heat of the steam corresponding to the pressure and q is the heat of the liquid of the condensate discharged.

The pipe or casing around the heater should be particularly well insulated to prevent introducing an error by radiation and convection loss.

In the *Thomas* electrical meter the air or gas is heated only a few degrees so that the error by radiation is negligible. The results obtained by this meter are generally conceded to be as accurate as it is possible to measure the flow.



Tempering Coil
2 Sects. Velocity 1000 ft.p.m. thro clear area (V.P.=.06")
Pr. loss=.15"
Suction Side
Pr. Loss=.25 V.P. in pipe or .015"
1000 ft.p.m. V.P.=.06"
Fan
1200 ft.p.m. V.P.=.09"
Discharge Side
Pr. Loss=.5 V.P. in pipe or .045"
Heater
4 Sects. Vel. 1200 ft.p.m. thro' clear area (V.P.=.09")
Pr. loss=.30"
Pr. Loss=.25 of V.P. in pipe or .06"
Duct System
Vel. 2000 ft.p.m. V.P.=.25"
Pr. Loss .25"
100'
Total or Dynamic Pr.
Static Pr. or Maintained Resistance
Barometric Pr. Line
H=1.07"
Hd=.905"
Suction Side
h_5 = pr. loss thro tempering coil = .15" water.
h_6 = " " in connection to fan = .015"
h_s = final velocity pr. at fan entrance = .06"
H_s = total suction pr.
= $(h_5+h_6)+h_s$ = .15+.015+.06 = .225"
Delivery Side
h_1 = pr. loss in duct system = .25"
h_2 = " " " connection of heater to duct = .06"
h_3 = " " " heater = .30"
h_4 = " " " connection of heater to fan = .045"
h_v = final velocity pressure = .25"
H_d = total delivery pressure.
= $(h_1+h_2+h_3+h_4)+h_v$ = .25+.06+.30+.045+.25
= .905"
Total head on fan $H = (H_s-h_s)+H_d$
= $(h_1+h_2+h_3+h_4+h_5+h_6)+h_v$
= .25+.06+.30+.045+.15+.015+.25 = 1.07" water
= Sum of friction pressure losses + final velocity pressure.

FIG. 36.

FRICTIONAL RESISTANCE AND TOTAL HEAD OR PRESSURE TO BE OVERCOME IN MOVING AIR

Delivery Side of Fan. Referring to Fig. 36, showing the delivery side of fan, it will be observed that the total and static pressures existing at various points are indicated by heights of water column in the U tubes connected to the Pitot and Piezometer tubes. *The loss in pressure between any two points in the system is always equal to the difference between the total pressures at these points, no matter how the velocity may change.* If the velocity remains constant, then the velocity pressure is constant and the difference in static pressures will also give the loss in pressure. This is only true so long as the area of duct remains constant.

In the pressure diagram given below the sketch, total pressure is shown by the solid line and static pressure by the dash line.

Let H_d = total pressure on delivery side of fan.
h_1 = pressure loss in duct.
h_2 = pressure loss in connection of heater to duct.
h_3 = pressure loss in heater.
h_4 = pressure loss in connection of fan to heater.
h_v = final velocity pressure.

Then $H_d = (h_1 + h_2 + h_3 + h_4) + h_v$ = (sum of pressure losses) + final velocity pressure.

Suction Side of Fan. The pressures on the discharge side of the fan, measured from atmospheric pressure, are positive, but when the air is drawn through a pipe by a fan both the static and total pressures existing in the pipe are below atmospheric, and are therefore negative. This is true of any system of ducts, etc., that may be connected to the suction side of a fan.

The *total pressure reading, in this case, is always less than the static pressure reading,* therefore when the static pressure is subtracted from the total pressure the result is positive, as it should be, for the velocity pressure.

The static or Piezometer tube on the suction side gives total pressure, that is, it includes the velocity pressure, and the Pitot tube gives the static pressure. The meaning of the readings being reversed in this case,

Let H_s = total suction pressure measured in inches of water.
h_5 = pressure loss in tempering coil.
h_6 = pressure loss in connection from tempering coil to fan.
h_s = final velocity pressure at fan entrance.

Then $H_s = (h_5 + h_6) + h_s$ = (sum of pressure losses) + final velocity pressure.

The total head or pressure H that must be supplied by the fan *is not the sum of total suction head and total delivery head,* as the total suction head includes the velocity head or pressure at the fan entrance.

The total pressure is therefore:

$$H = (H_s - h_s) + H_d$$

but $H_s = h_5 + h_6 + h_s$ and $H_d = h_1 + h_2 + h_3 + h_4 + h_v$

or $H = (h_1 + h_2 + h_3 + h_4 + h_5 + h_6) + h_v$

= (sum of all pressure losses) + final velocity pressure.

Power Required to Move Air. Let T, P, or H = total head or pressure to be overcome, measured in inches of water.

= sum of pressure losses + final velocity pressure.

P = unit pressure lb. per sq. ft.

$$= 62.4 \times \frac{H}{12} = 5.2\ H.$$

Q = quantity of air flowing cu. ft. per min.
V_m = velocity of air ft. per min.
A = area of pipe sq. ft.
$P\,A$ = total pressure lb.
$P\,A\,V_m$ = work, ft. lb. per min.

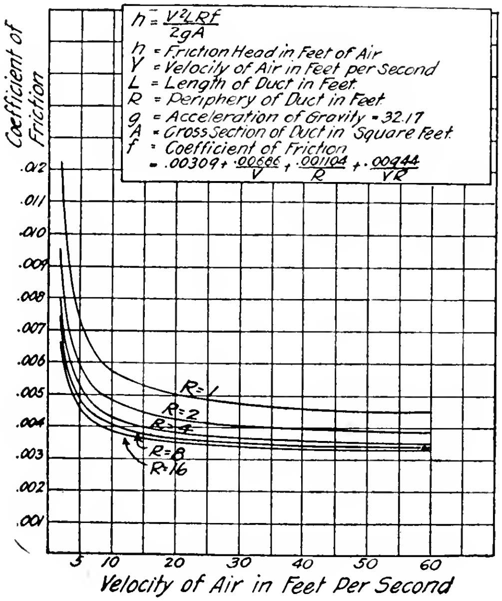
Then air horsepower $H.\,P._a = \dfrac{P\,A\,V_m}{33000} = \dfrac{5.2\,H\,Q}{33000}$.

If E = efficiency of fan or blower,

$$= \frac{H.\,P.\text{ output}}{H.\,P.\text{ input}} = \frac{\text{Air } H.\,P.}{\text{Brake } H.\,P.}$$

Then the brake horsepower of the fan, $D.\,H.\,P. = \dfrac{H.P._a}{E} = \dfrac{5.2\,H\,Q}{E \times 33000}$.

The efficiency of fans is quite variable, depending upon a number of items, each of which affects the efficiency as will be shown later under "Performance and Rating of Fans."



Coefficient Friction

FIG. 37.

For enclosed fans when *operated only at their tabulated capacity and pressure rating* the following efficiencies may be used:

Steel plate fans $E = 0.46$ for a ratio $\dfrac{\text{total pr.}}{\text{static pr.}} = 1.36$.

Multibladed fans $E = 0.61$ for a ratio $\dfrac{\text{total pr.}}{\text{static pr.}} = 1.15$

These efficiencies refer particularly to the fan Tables 23 and 24.

It should be noted that fans are rated at or near their point of highest efficiency, that is, when the ratio of the total to static pressure at the fan outlet is some fixed amount. When this ratio is deviated from the efficiency rapidly falls off as shown by the test curve (Fig. 46).

Loss of Pressure in Air Ducts. The general form of the expression for the loss of head measured by the height of a column, in ft., of the medium flowing in a pipe or duct is:

$$h_x = f\frac{L\,R\,v^2}{A\;2g} \quad \text{or } f\frac{L\,R}{A} \times \text{velocity head or pressure,} \qquad (1)$$

in which

h_x = head loss measured in ft. of air column
L = length of the pipe or duct in ft.
R = perimeter of the pipe or duct in ft.
A = area of the pipe or duct in sq. ft.
$L\,R$ = area of the rubbing surface in sq. ft.
f = coefficient of friction.
v = velocity of flow (average over the cross section) in ft. per sec.
g = 32.16.
$\frac{v^2}{2g}$ = velocity head or pressure.

Experiments on the flow of air in *smooth* sheet steel ducts, by *Reitschel* and others, varying in size from 12″ to 48″ diam. and with velocities ranging from 1000 to 2400 ft. per minute show that the average value of f is 0.0037 (Fig. 37). The value of f that is used by various manufacturers in the calculation of published tables and diagrams is approximately 0.0062 and is undoubtedly too high.

If the measurements for head lost are made or stated in inches of water, formula (1) becomes

$$h_x = \frac{h \times k}{12\,d} = f\,\frac{L\,R}{A} \times \frac{v^2}{2g}$$

or

$$h = \frac{12\,d}{K} \times f \times \frac{L\,R}{A} \times \frac{v^2}{2g} \qquad (2)$$

in which

h = head lost measured in inches of water.
d = density of the air flowing (wt. per cu. ft.).
= 0.075 for 70° F.
K = density of the water in the manometer.
= 62.4 for 70° F.

For a temperature of 70° F., (2) becomes

$$h = 0.00022\,f \times \frac{LR}{A}\,v^2 \qquad (3)$$

and for round ducts 100 ft. long

$$h = 0.088\,f\,\frac{v^2}{D} \qquad (4)$$

in which D = diameter of the duct in ft.

$$\text{If } f = 0.00372 \text{ then } h = 0.000327\,\frac{v^2}{D} \text{ or } h = \frac{0.021}{D} \times \frac{v^2}{2g} \qquad (5)$$

$$f = 0.0062 \text{ then } h = 0.000545\,\frac{v^2}{D} \text{ or } h = \frac{0.035}{D} \times \frac{v^2}{2g} \qquad (6)$$

If Q = cu. ft. of air flowing per minute $= 60\ A\ v$ or

$$v = \frac{Q}{60\ A}$$

Substituting the value of v in (3) we have

$$h = 0.00022\, f\, \frac{L\ R\ Q^2}{3600\ A^3} \qquad (7)$$

or

$$Q = 4045\ \sqrt{\frac{h}{Lf}} \times \sqrt{\frac{A^3}{R}} \qquad (8)$$

From this equation it is evident that the head lost for a given value of Q will be the same for either round or rectangular ducts so long as the value of $\sqrt{\frac{A^3}{R}}$ remains the same in both cases.

For a value of $f = 0.0037$ and $L = 100$ ft.,

$$Q = 6633\ \sqrt{h} \times \sqrt{\frac{A^3}{R}} \qquad (9)$$

For round ducts $\sqrt{\frac{A^3}{R}} = 0.393\ \sqrt{D^5}$ in which D = the diameter of the duct in ft.,

then

$$Q = 2607\ \sqrt{h\ D^5} \qquad (10)$$

for smooth sheet steel ducts with no allowance for roughness of joints, etc.

For the value of $f = 0.0062$ as ordinarily used in the calculation of manufacturers' tables and charts.

$$Q = 5140\ \sqrt{h} \times \sqrt{\frac{A^3}{R}} \qquad (11)$$

which for round ducts reduces to

$$Q = 2020\ \sqrt{h\ D^5}. \qquad (12)$$

The accompanying chart (Fig. 38) was calculated by means of equations (9) and (10). It was thought best to plot the chart for a value of $f = 0.0037$ corresponding to smooth ducts and in using the chart to add a percentage which in the judgment of the designer would cover the increased friction due to rough joints, poor alignment, etc.

In this connection the authors would recommend *the addition* of 50 per cent to the friction loss as read from the chart for the average well constructed duct layout when proper allowance has been made for the ells or turns, loss through the register grills, etc. The loss in concrete ducts may be assumed as 1½ times and in brick ducts or flues as 2 times the loss in sheet steel ducts of the same dimensions. The chart (Fig. 38) was plotted for a temperature of 70° F.; the pressure loss for any other temperature, for the same size duct and volume flowing, will be equal the pressure loss from chart times the density of the air divided by 0.075.

Explanation of Chart. (Fig. 38.) Locate the value of Q on the right hand side of chart, pass horizontally to the left to the intersection with the vertical pressure loss per 100 ft. for which it is desired to design the duct. The diagonal line passing through this intersection gives the diameter of the round duct required. The intersection of this diameter line with the line marked "$\sqrt{\frac{A^3}{R}}$ for round ducts" gives this value on the scale at the top of the chart. The equivalent dimensions of a rectangular duct for the same value of Q and h may be read from the

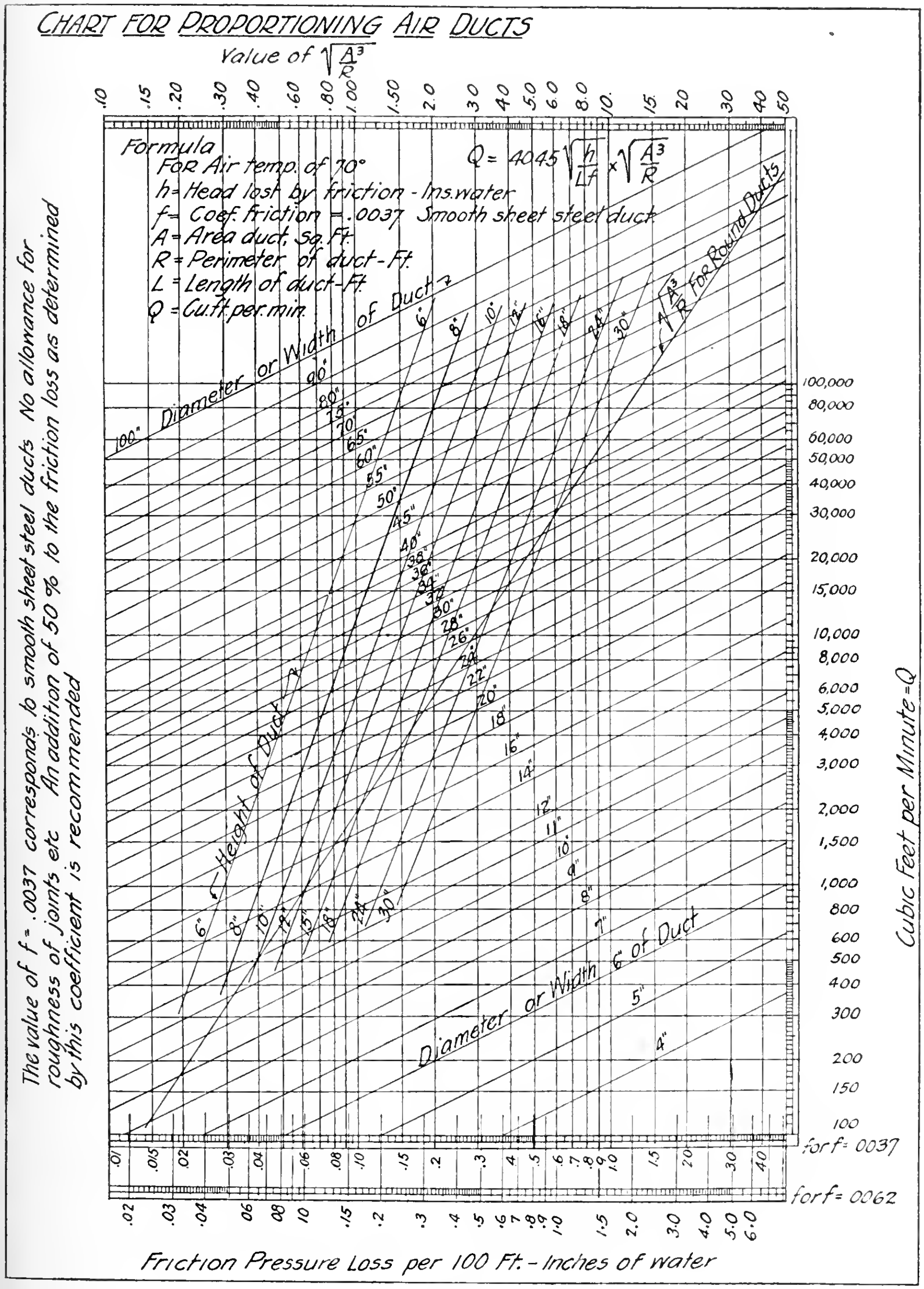


CHART FOR PROPORTIONING AIR DUCTS
Value of $\sqrt{\frac{A^3}{R}}$
Formula
For Air temp. of 70°
h = Head lost by friction - Ins. water
f = Coef. friction = .0037 Smooth sheet steel duct
A = Area duct, Sq. Ft.
R = Perimeter of duct - Ft.
L = Length of duct - Ft.
Q = Cu. ft. per min.
$Q = 4045\sqrt{\frac{h}{Lf}} \times \sqrt{\frac{A^3}{R}}$
Diameter or Width of Duct
Height of Duct
$\sqrt{\frac{A^3}{R}}$ for Round Ducts
Diameter or Width 6" of Duct
The value of f = .0037 corresponds to smooth sheet steel ducts. No allowance for roughness of joints etc. An addition of 50% to the friction loss as determined by this coefficient is recommended
Cubic Feet per Minute = Q
for f = .0037
for f = .0062
Friction Pressure Loss per 100 Ft. - Inches of water

FIG. 38.

intersection of the vertical with the diagonals marked "height of duct." The intersection of the diagonal line marked "diameter or width of duct" with this point gives the required width.

Example. Required the pressure loss in a 24″ round duct 75 ft. long when conveying 6000 cu. ft. of air per min. measured at 70° F. Locate 6000 on the right hand side of the chart, pass horizontally to the left to the intersection with the diagonal 24″ line, then down and read 0.175″ for the loss per 100 ft. For 75 ft. the loss is therefore equal to $75/100 \times 0.175''$ or 0.131″ for a smooth duct; adding 50 per cent as an allowance for rough joints, etc., we have 0.131×1.50 or 0.196″ for the probable loss.

The value of A^3/R for the 24″ round duct as read at the top of the chart is 2.2.

If the duct is to be rectangular for the same pressure loss, and having a height of 15″, then the width will be 34″.

Table 16, following, giving the rectangular equivalent of circular ducts, is frequently quoted in the literature of fan manufacturers:

TABLE 16

Side Rectangular Duct	4	6	8	10	12	14	15	16	18	20	22	24
	Equivalent Diameters											
3	....	....	....	....	....	....	....	....	....	....	....	....
4	4.4	....	....	....	....	....	....	....	....	....	....	....
5	4.9	....	....	....	....	....	....	....	....	....	....	....
6	5.4	6.6	....	....	....	....	....	....	....	....	....	....
7	5.8	7.0	....	....	....	....	....	....	....	....	....	....
8	6.1	7.6	8.8	....	....	....	....	....	....	....	....	....
9	6.5	8.0	9.3	....	....	....	....	....	....	....	....	....
10	6.8	8.4	9.8	11.0	....	....	....	....	....	....	....	....
11	7.1	8.8	10.2	11.5	....	....	....	....	....	....	....	....
12	7.4	9.2	10.7	12.0	13.2	....	....	....	....	....	....	....
13	7.6	9.6	11.1	12.5	13.7	....	....	....	....	....	....	....
14	7.6	9.9	11.5	12.9	14.3	15.4	....	....	....	....	....	....
15	8.2	10.2	11.9	13.4	14.7	16.0	16.5	....	....	....	....	....
16	8.4	10.5	12.3	13.8	15.2	16.5	17.1	17.6	....	....	....	....
17	8.6	10.8	12.6	14.2	15.7	17.0	17.6	18.2	....	....	....	....
18	8.9	11.1	13.0	14.6	16.1	17.4	18.1	18.7	19.8	....	....	....
19	9.1	11.4	13.3	15.0	16.5	17.9	18.6	19.2	20.4	....	....	....
20	9.3	11.6	13.6	15.4	17.0	18.4	19.0	19.7	20.9	22.0	....	....
22	9.7	12.1	14.2	16.1	17.8	19.2	19.9	20.6	21.9	23.1	24.2	....
24	10.0	12.6	14.8	16.8	18.5	20.0	20.8	21.5	22.8	24.0	25.2	26.4
26	10.4	13.1	15.4	17.3	19.2	20.8	21.6	22.3	23.8	25.1	26.3	27.5
28	10.8	13.5	15.9	18.0	19.8	21.5	22.4	23.1	24.6	26.0	27.3	28.5
30	11.0	13.9	16.4	18.5	20.5	22.2	23.1	23.9	25.4	26.8	28.2	29.5
32	11.3	14.3	16.9	19.1	21.1	22.9	23.8	24.6	26.2	27.7	29.1	30.5
34	11.6	14.7	17.3	19.6	21.6	23.5	24.4	26.3	26.9	28.5	30.0	31.3
36	11.9	15.1	17.7	20.1	22.2	24.2	25.1	26.0	27.7	29.3	30.8	32.2
38	12.2	15.4	18.2	20.6	22.8	24.8	25.8	26.7	28.4	30.0	31.5	33.1
40	12.5	15.7	18.6	21.1	23.3	25.4	26.4	27.3	29.1	30.8	32.4	33.9
42	12.7	16.1	19.0	21.6	23.8	25.9	26.9	27.9	29.8	31.4	33.0	34.5
44	13.0	16.4	19.4	22.0	24.3	26.5	27.5	28.5	30.3	32.1	33.7	35.3
46	13.3	16.7	19.8	22.4	24.8	27.0	28.1	29.1	31.0	32.8	34.6	36.2
48	13.5	17.0	20.1	22.8	25.2	27.5	28.6	29.6	31.6	33.4	35.2	37.0
50	13.7	17.3	20.4	23.2	25.7	28.0	29.2	30.3	32.2	34.1	35.9	37.6
52	13.9	17.6	20.8	23.6	26.2	28.5	29.6	30.7	32.9	34.7	36.5	38.3
54	14.1	17.9	21.1	24.0	26.6	29.0	30.1	31.2	33.4	35.3	37.2	38.9
56	14.3	18.2	21.5	24.4	27.0	29.5	30.6	31.7	33.9	35.9	37.8	39.6
58	14.6	18.4	21.8	24.7	27.4	30.0	31.1	32.2	34.4	36.4	38.4	40.3
60	14.7	18.7	22.1	25.1	27.8	30.5	31.6	32.7	34.9	37.1	39.1	40.9
62	15.0	19.0	22.4	25.5	28.2	30.9	32.1	33.2	35.4	37.7	39.6	41.6
64	15.1	19.2	22.7	25.9	28.6	31.3	32.6	33.7	35.9	38.2	40.2	42.2
66	15.3	19.5	23.0	26.2	29.0	31.7	33.0	34.2	36.4	38.7	40.8	42.8
68	15.5	19.7	23.3	26.5	29.4	32.1	33.4	34.7	36.9	39.2	41.4	43.4

Various Pressure Losses. The *pressure loss* occasioned by ells, enlargements of cross section, entrance, register grills, etc., is ordinarily stated by the following expression as a function of the velocity head:

$$h = K \times \frac{V^2}{2g} \text{ or } K \times \text{the velocity pressure (V. P.)}$$

in which h = the loss of pressure measured in inches of water.
K = a coefficient determined by experiment.
V = velocity in ft. per sec.

The values of K for *ells, entrance, enlargement,* etc., are given by Fig. 39. Pressure loss through heaters and air washers is given by Tables 17 and 18.

Pressure losses are also designated by the equivalent length of straight pipe which will give the same loss. Thus for an ell, when the ratio of the throat radius to the diameter is equal to

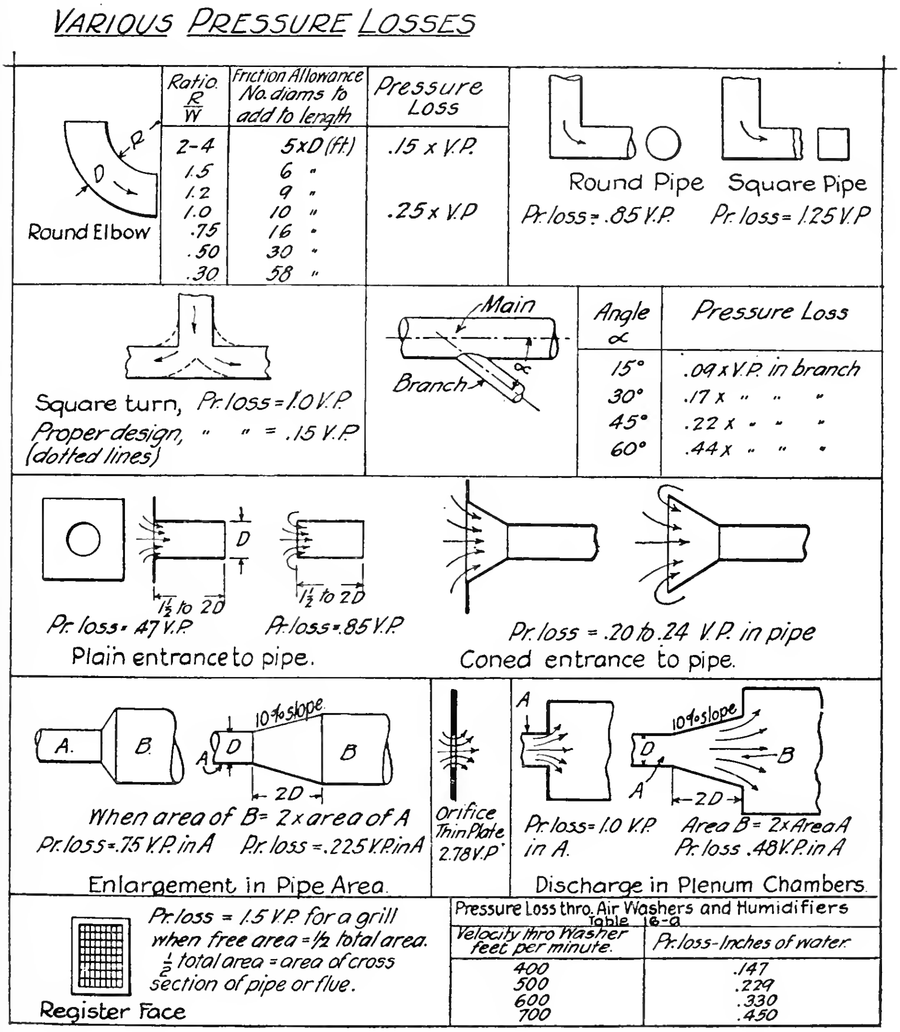


FIG. 39.

2 to 4, the pressure loss is 0.15 × the velocity pressure or it may be stated as equal to the pressure loss in a length of pipe, of the same diameter, 5 × diam. in ft. In other words, this equiva-

lent length as an allowance for the ell is added to the measured length of pipe to obtain the total length for estimating the friction pressure loss in the duct.

A numerical example showing the application of the friction pressure loss data in determining the total head on a fan is given by Fig. 36. It is recommended that the velocity through the heater be limited to such an amount that the pressure loss in the heater will be not greater than one half the total fan pressure required.

TABLE 17

FRICTION EFFECT THROUGH HOT BLAST HEATERS

(B. F. Sturtevant Co.)

Friction in inches of water. Volumes of air and friction effect taken at 65° F.
Outside Diam. 1″ Pipe = 1.281″. Outside Diam. 1¼″ Pipe = 1.660″

Velocity Through Free Area, Feet per Min.	Number of Rows in Heater	1″ Pipe 2⅛″ Centers	1¼″ Pipe 2⅝″ Centers	1″ Pipe 2⅝″ Centers	1″ Pipe 2¾″ Centers
600	8	0.07	0.07	0.05	0.05
	16	.11	.11	.07	.07
	24	.15	.16	.10	.10
	32	.19	.21	.13	.13
900	8	.15	.16	.11	.11
	16	.24	.26	.17	.16
	24	.34	.37	.23	.21
	32	.44	.48	.29	.27
1200	8	.26	.28	.20	.19
	16	.43	.47	.30	.28
	24	.60	.66	.41	.38
	32	.78	.85	.52	.48
1500	8	.41	.44	.31	.29
	16	.68	.73	.47	.45
	21	.95	1.03	.64	.60
	32	1.22	1.33	.81	.76
1800	8	0.59	0.63	.44	.42
	16	0.98	1.05	.69	.64
	24	1.36	1.48	.93	.86
	32	1.75	1.91	1.17	1.08
2100	8	0.80	0.86	0.61	0.58
	16	1.33	1.44	0.94	0.88
	24	1.85	2.02	1.27	1.18
	32	2.39	2.61	1.60	1.48

TABLE 18

FRICTION OF AIR THROUGH VENTO HEATERS

Friction Loss—in Inches of Water—Due to Air Passing Through Vento Stacks
Regular Section
Standard 5-Inch Spacing of Loops

Velocity Feet per Min.	1 Stack	2 Stacks	3 Stacks	4 Stacks	5 Stacks	6 Stacks	7 Stacks
600	0.021	0.040	0.058	0.076	0.094	0.112	0.130
700	.028	.054	.079	.105	.130	.155	.180
800	.037	.070	.103	.135	.167	.200	.232
900	.047	.088	.129	.170	.211	.252	.293
1000	.059	.109	.160	.211	.262	.313	.364
1100	.071	.132	.193	.255	.316	.377	.438
1200	.084	.157	.230	.303	.376	.449	.522
1300	.099	.185	.271	.356	.442	.528	.614
1400	.115	.214	.314	.414	.513	.612	.712
1500	.132	.246	.360	.474	.588	.702	.816
1600	.150	.280	.410	.540	.670	.800	.930
1700	.169	.316	.463	.609	.756	.903	1.049
1800	.190	.354	.518	.683	.848	1.012	1.177

Total Resistance in Practice. The total head or pressure H against which the fan must operate in practice is ordinarily about as follows:

0.30″ Public building ventilation without heaters, system well proportioned.

0.60″ Public building ventilation with complex system and without heaters.

0.60–0.90″ Public building heating without air washers.

1.15″ Public building heating with air washers.

0.75″ – 1.25″ Factory heating systems, without air washers.

The corresponding static pressure or maintained resistance rating for the fan is given later in the text.

Allowable Velocities in Ducts and Flues. Recommended velocities through heaters for various classes of work have already been stated. Modern practice limits the velocities in the ducts, flues and registers, in order to prevent excessive friction loss, in accordance with the figures stated by Table 19, following.

In public building work the air should be delivered to the room at such velocity that will secure its movement to the desired points in the room without objectionable drafts or noise in passing through the register grills.

TABLE 19

ALLOWABLE VELOCITIES IN HOT BLAST SYSTEMS

	Allowable Velocity, Feet per Minute
Public Buildings	
Through free area of wall registers	400– 500
Through free area floor registers	200– 300
Vertical flues to registers	600– 750
Connections to base of flues	800–1000
Main horizontal distributing ducts	1500–2500
Manufacturing Plants	
In plants where the occupation is more or less sedentary the employe sits all day feeding automatic machinery, as in glove and shirt manufacturing, etc.	
Main ducts	1200–1500
Branches	600– 900
In plants where the employe stands all day as in machine shops, foundries, etc.	
Main ducts	1500–2400
Branches	900–1500

The velocity through the fan outlet, under ordinary conditions that obtain in heating work, varies from 1500 to 2500 feet per minute.

VENTILATING FANS

Steel Plate Fan. The standard type of fan that has been used for a number of years in hot blast work is known as the Steel Plate Fan, the construction of the wheel being shown by Fig. 40. As the name implies, the wheel and casing of this fan are constructed of steel plate and light structural sections, the wheel having eight to twelve blades straight or slightly curved at the periphery in the opposite direction to the rotation. Table 21 gives the dimensions of steel plate fans of several manufacturers and will be found of use in making provision for the space necessary for this part of the system.

Steel plate fans are designated by number, this number being the approximate height of the fan casing in inches, corresponding to dimension A of Fig. 49.

Multibladed Fan. A new type of fan has recently entered the field of heating and ventilating, and on account of its higher efficiency, quieter running, and occupying less space, for the same capacity, than the steel plate fan, it is rapidly supplanting the latter. The higher efficiency is accounted for by the material reduction of the air resistance or pressure head loss by friction through the fan, due to the shorter blades and the larger inlet which is practically the same diameter as the wheel. This fan deserves more than passing mention as it represents perhaps the greatest single improvement ever made in the design of a centrifugal fan. This fan

is the invention of *S. C. Davidson*, of *Belfast, Ireland*, and is known as the *Sirocco* fan or blower. (Fig. 41.) The fan wheel has from forty-eight to sixty-four blades of shallow depth, approximately 1/16 the fan diameter, the blades being curved both at the inlet and periphery of the wheel in the direction of rotation. The width of wheel is approximately equal to 6/10 the diameter.

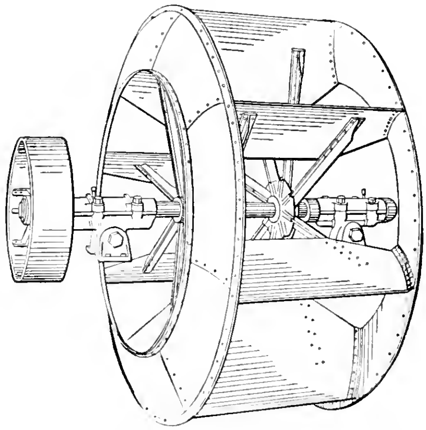


FIG. 40. STANDARD STEEL PLATE FAN WHEEL.

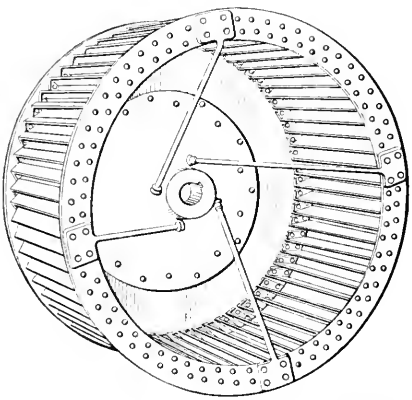


FIG. 41. SIROCCO WHEEL—TURBINE TYPE IMPELLER.

This fan is now manufactured by the *American Blower Co.* A modification of the *Sirocco* fan is manufactured by the *B. F. Sturtevant Co.,* and termed the *Multivane* fan (Fig. 42), and by the *Massachusetts Fan Co.,* under the name of *Squirrel Cage.*

The *Conoidal* fan, manufactured by the *Buffalo Forge Co.,* is a multibladed type of fan having a wheel in the shape of a frustrum of a cone as shown by Fig. 43.

The accompanying diagrams, Fig. 44, show various alternative arrangements of the position of driving pulley and direction of outlet of fans.

The terms "Right Hand" and "Left Hand" refer to the position of the outlet relative to a person facing the pulley or driving side of the fan; exactly opposite for *B. F. Sturtevant Co.'s* fans.

The fan wheel is right or left hand, according to the direction of rotation when viewed from pulley side. The direction of rotation is indicated in the above diagrams by an arrow.

PERFORMANCE AND RATING OF FANS

A clear understanding of the method employed in arriving at the rating given by fan manufacturers to their product is essential in order to make intelligent use of these data.

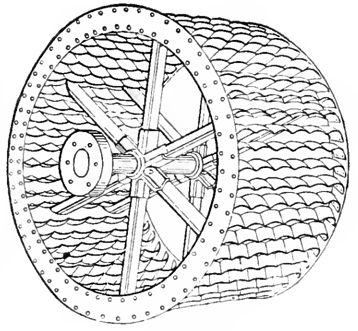


FIG. 42.
MULTIVANE FAN WHEEL.

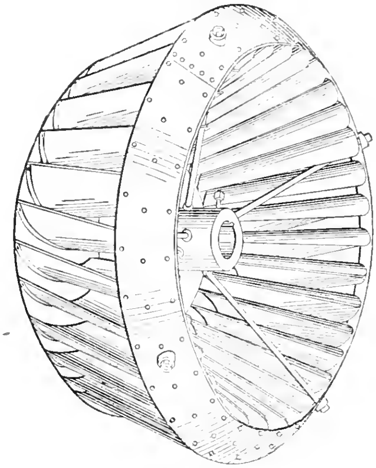


FIG. 43.
NIAGARA CONOIDAL BLAST WHEEL.

If we expect a fan, or in fact any other mechanical apparatus, to perform its guaranteed or tabulated duty *we must reproduce the conditions in practice, as to pressures and volume, under which the fan was originally rated.* Unless this is fully realized, disappointing results are quite likely to occur.

The variable relation existing between the speed, pressure generated and volume delivered in both fans and centrifugal pumps have frequently been the cause of grave errors in both the design and operation of systems employing this class of apparatus.

Rating Tests. A method employed in the rating test of fans is as follows: (Fig. 45). The fan is set up with a free and unobstructed inlet and to the outlet is attached a pipe the area of which is the same as the area of the fan outlet and length equal to approximately 20 diameters. A frame is attached to the end of the discharge pipe arranged to receive shutters having various size openings. These openings are 10, 20, 30, 40, 50, 60, 70, 80 and 90 per cent of the pipe area.

The object of these shutters is to provide a variable resistance head or pressure for the fan. A combined piezometer and Pitot tube is inserted midway in the length of pipe to register the static, total and velocity pressures.

When the end of the pipe is full open (100 per cent opening) the static pressure is practically 0 and would be zero if it were not for the small amount of friction in the duct. The total pressure registered by the Pitot tube is in this case also the velocity pressure neglecting the small difference noted above.

If the opening is now partially closed, the fan operating at the same speed as before, the effect is equivalent to adding resistance in the form of a heater system of air ducts, etc., to the fan. The piezometer will now register a reading and the difference between the total and static pressure gives the velocity pressure. It will be observed that the velocity pressure has changed, becoming less as the area of the pipe opening is reduced, until the static and total pressure is the same when the opening is entirely closed, the velocity pressure being zero.

For *economical operation*, a fan, like other power transformation apparatus, should be *rated and used at or near its capacity at highest efficiency.* It is the object of the rating test to determine the capacity of the fan for its highest efficiency when operating at various pressures.

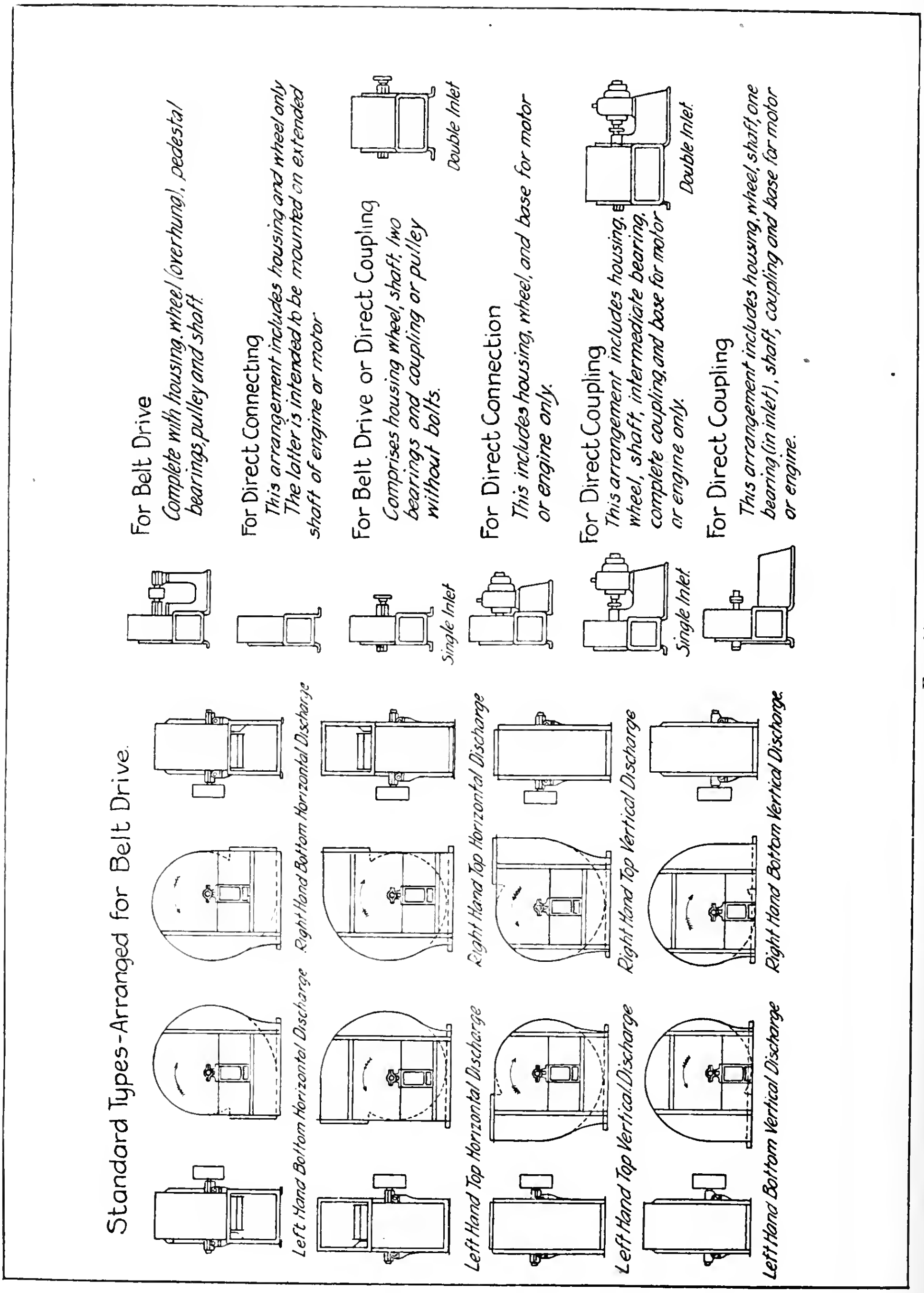

Standard Types-Arranged for Belt Drive.
Left Hand Bottom Horizontal Discharge
Right Hand Bottom Horizontal Discharge
Left Hand Top Horizontal Discharge
Right Hand Top Horizontal Discharge
Left Hand Top Vertical Discharge
Right Hand Top Vertical Discharge
Left Hand Bottom Vertical Discharge
Right Hand Bottom Vertical Discharge
For Belt Drive
Complete with housing, wheel (overhung), pedestal bearings, pulley and shaft.
For Direct Connecting
This arrangement includes housing and wheel only. The latter is intended to be mounted on extended shaft of engine or motor
For Belt Drive or Direct Coupling
Comprises housing wheel, shaft, two bearings and coupling or pulley without bolts.
Single Inlet
Double Inlet
For Direct Connection
This includes housing, wheel, and base for motor or engine only.
For Direct Coupling
This arrangement includes housing, wheel, shaft, intermediate bearing, complete coupling, and base for motor or engine only.
Single Inlet
Double Inlet
For Direct Coupling
This arrangement includes housing, wheel, shaft, one bearing (in inlet), shaft, coupling and base for motor or engine.

Fig. 44.

From past experience, we know that a single stage fan will not be used for static pressures exceeding several inches of water so that the tests will be limited to a maximum total pressure of, say, 5" water static pressure and that a table giving the capacity and power required for

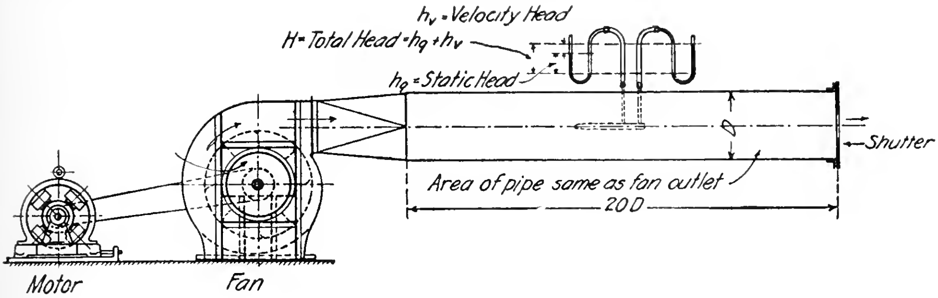


FIG. 45.

static pressures up to 5", varying by approximately ¼ inch, will fulfil all practical requirements.

The fan is first operated at constant speed (about 2000 ft. per min. tip speed) and the opening at the end of the pipe varied by the shutters. Readings are taken and computations made in accordance with the following table for each opening. It is found that as the opening is varied the relation between the total static and velocity pressure varies as does also the efficiency:

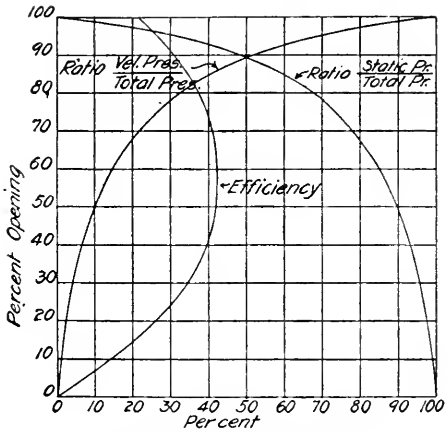
TABLE 20

Items	Per cent Opening								
	10	20	30	40	50	60	70	80	90
Total Pr. =									
Static Pr. =									
Vel. Pr. =									
Ratio $\frac{\text{total Pr.}}{\text{static Pr.}}$ =									
Velocity (V) =									
Capacity ($A \times V$) =									
Air H. P. = $\frac{60 \times 5.2\ T.\ P. \times AV.}{33000}$ =									
Brake H. P. =									
Efficiency (E) $\frac{\text{Air H. P.}}{\text{Brake H. P.}}$ =									

Curves similar to those shown by Fig. 46 may be constructed from the foregoing data.

These curves show that for fans of standard construction the maximum efficiency is obtained when the opening is approximately 70 per cent for steel plate and 50 per cent for *Sirocco* type fans, corresponding to a ratio of total to static pressure of 1.36 and 1.15 respectively. The opening which gave the highest efficiency is now used and the fan speed varied so that the various pressures desired are obtained and a fan table similar to Tables 23 and 24 constructed. In choosing a fan two items must be known; that is, the volume to be delivered and the static or total pressure against which the fan is to operate. Having located these in the table the corresponding speed and brake horsepower required are obtained.

Tables 23 and 24 following give the static pressure or maintained resistance rating of the fans. To obtain the corresponding total or dynamic pressure rating add to the static pressure the velocity pressure corresponding to the velocity of the air through the fan outlet.



Typical Performance Curves
Steel Plate Fan

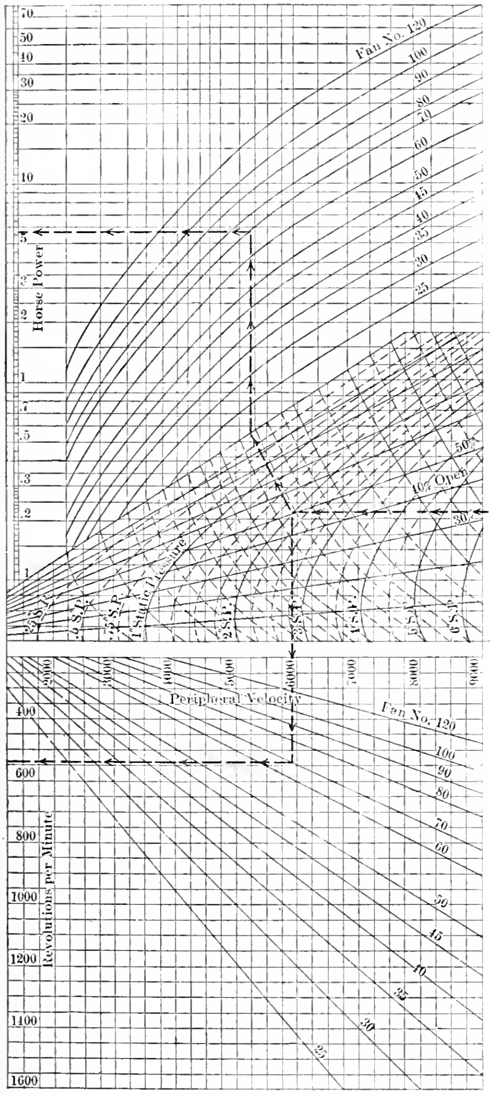
FIG. 46.

TABLE 21
DIMENSIONS OF STEEL PLATE FANS
Full Housing

No. Fan	Mfg'r.	A	Wheel Dia.	Wheel Width	Inlet K	Outlet H	Outlet I	B	C	D	E	Pulley Dia.	Pulley Face	P	Area Outlet, Sq. In.	Wgt., Lbs.
70	1	70	42	17¼	26	24½	24½	33½	23¾	24½	33	16	4	23½	324	1000
	2	71	42	18½	27	25	25	34¼	24½	27¼	33¼	16	4	23⅛		
	3		42	16¾	30	25¾		34¼	28¾	27¼	39¼			23½		
80	1	80	48	18¼	30	27	27	38½	26½	27	38	18	4	24½	462	1200
	2	80½	48	19½	30½	27	27	39	26½	31	37½	18	4	24⅛		
	3	79½	48	18¾	34	24¾	28½	39	30½	31	36½			26¼		
90	1	89	54	21	34	30½	30½	42¾	30¼	30½	42	20	4	26½	600	1450
	2	90	54	22¾	34½	31	31	43¾	30¼	34¾	41¾	20	5	26⅝		
	3	94	54	22¾	38⅞	28¾	30½	44	34¼	34	45					
100	1	99	60	23¾	38	34½	34½	47½	33½	34½	47	22	6	30½	729	2000
	2	100	60	24⅞	38	34	34	48½	32¾	38½	46½	22	6	30⅜		
	3	105	60	24¼	42⅜	32¾	32½	49	36⅛	37	50			31¾		
110	1	108	66	26	42	37½	37½	52	37¼	37½	51	24	6	32	930	2400
	2	109½	66	27	42	37½	37½	53¼	35¾	42¼	50¾	24	6	32⅛		
	3	110½	66	29	47	36¾	36½	53½	40½	40½	50½			33¾		
120	1	117	72	29⅜	46	41½	41½	56¾	41¼	41½	55	26	6	35½	1188	3000
	2	119½	72	30	46	41	41	58	39½	46	55¾	26	6	35⅜		
	3	120	72	35¼	52¼	42⅞	42⅝	58	41½	44	55			34½		
140	1	136	84	34⅜	53	48	48	66	48	48	64	30	8	39½	1406	3800
	2	138½	84	35	53½	48	48	67½	45½	53½	64	30	7	39⅝		
	3	141	84	40¼	60½	48⅞	48⅝	68	45½	50	64			38½		
160	1	154	96	38¼	60	54	54	75¼	54	54	72	32	8	43	1722	4800
	2	157½	96	38	60	54	54	77	51½	61	72½	32	8	43⅝		
	3	159	96	39	68½	48⅞	54⅝	77	49½	57	72			39¾		
180	1	174	108	42⅞	68	60	60	84½	60½	60	82	36	8	46	2304	6000
	2	178	108	42	68	60	60	86½	57½	68½	81½	36	8	46⅝		
	3	177	108	44¼	77	55	60¾	87	57½	65	80			42¾		
200	1	192	120	47	76	66	66	93¾	66½	66	90	40	10	52½	2916	7400
	2	196	120	46	76	66	66	96	63½	76	90	40	10	52⅛		
	3	196	120	50¼	85½	61	72¾	96	68½	72	88			47		

1—*American Blower Co.*
2—*Massachusetts Fan Co.*
3—*B. F. Sturtevant Co.*
NOTE.—Width of wheel refers to width at periphery.
Letters refer to Fig. 49.

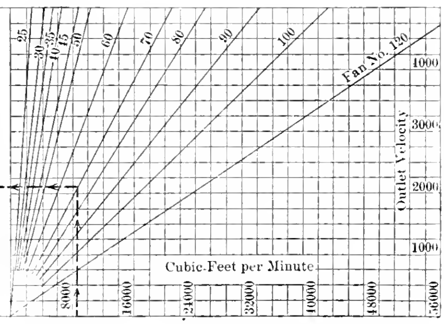
The sizes of the outlets are given by Tables 21 and 22. The velocity through the fan outlet is equal to the volume delivered divided by the area of outlet in sq. ft. Having thus obtained the velocity, the corresponding velocity pressure may be read direct from the curve (Fig. 31).



EXAMPLE IN USE OF CHART AS ILLUSTRATED WITH HEAVY DOTTED LINES AND ARROWS.

Assumptions: 8,800 cubic feet per minute, fan No. 70, static pressure 2″.

Method: Project the intersection of lines 8,800 C. F. M. and fan No. 70 to outlet velocity scale, read 2,100 feet per minute,



project same to 2″ static pressure, read 50% ratio of opening, project this intersection to peripheral velocity scale, read 5,925 feet per minute, continue to fan No. 70 line, thence to R.P.M. scale, read 540 R.P.M.; return to previous intersection on 2″ static pressure line, proceed parallel to horsepower lines to fan No. 70 line, project this intersection to horsepower scale and read 5.75 brake horsepower.

Based on 70° air. Altitudes up to 1,000′.

FIG. 47. PERFORMANCE CURVES AMERICAN STEEL PLATE FANS.

Example. The total pressure rating for a No. 7 Sirocco fan rated at 1″ static pressure when delivering 15,070 cu. ft. per min. (Table 24) is found as follows:

Area of outlet from Table 22 is 9.7 sq. ft.

Velocity 15,070/9.7 = 1553 ft. per min. The corresponding velocity pressure from diagram Fig. 31 is 0.15″.

The total pressure rating is therefore 1. + .15 or 1.15″.

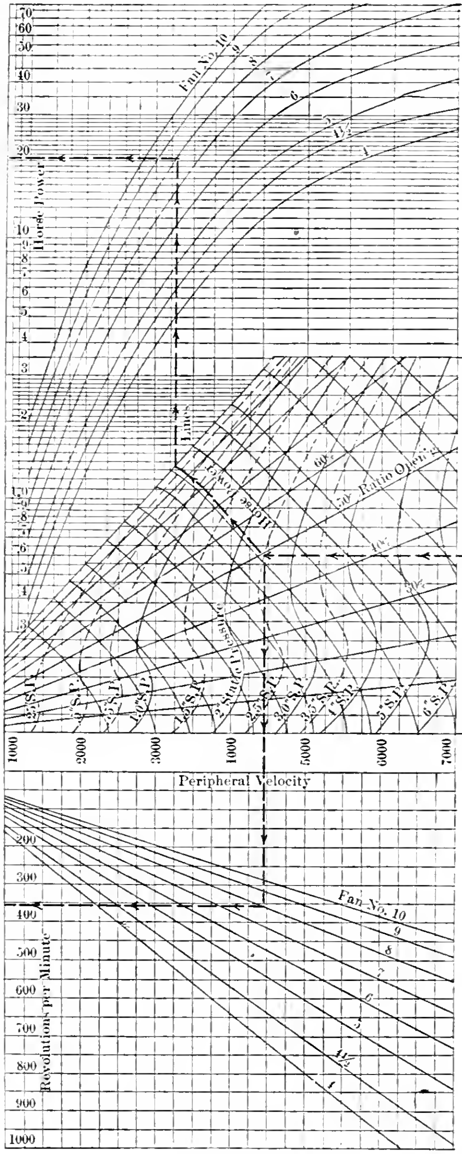
The ratio of total to static pressure is a constant for any capacity table based on tests with a constant opening, as previously shown. Therefore to determine the total pressure rating for

EXAMPLE SHOWING APPLICATION OF CURVES.

Problem. Determine size of fan, revolutions per minute and brake horsepower required to deliver 30,000 cubic feet per minute against a static pressure of 2.5″ W. G.

NOTE: For highest efficiency select a fan working between 45 and 50 per cent ratio of opening.

Solution. Locate 30,000 on the C. F. M. scale and project vertically upward from this point to the intersection of that fan working nearest 50% ratio opening at 2.5″ S. P., which in this case is fan No. 8 (follow dash lines on curves). From this point project horizontally to intersect 2.5″ S. P. curve and from thence upward parallel to horsepower lines to intersect fan No. 8 at which point read 20.2 D. H. P. on horsepower scale to left. From same point on S. P. curve project vertically downward to intersect fan No. 8 reading 355 R.P.M. on scale to left.



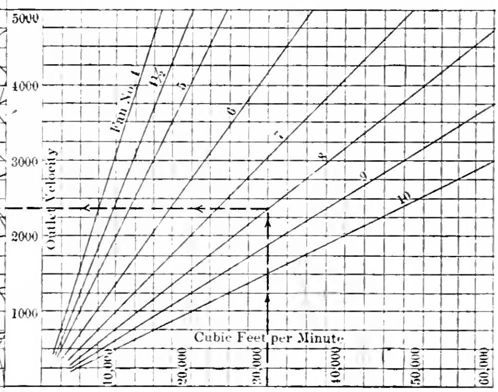
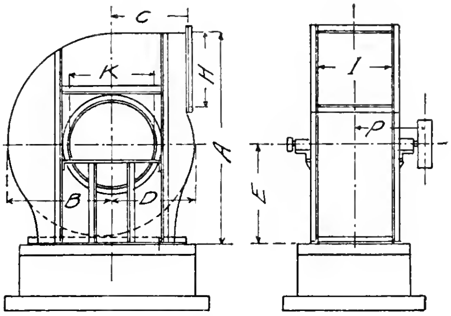


FIG. 48. PERFORMANCE CURVES AMERICAN SIROCCO FANS.

the steel plate fans (Table 23), multiply the static pressure by 1.36 and for the Sirocco fans (Table 24), multiply the static pressure by 1.15.

Example. The total pressure to be overcome in a certain system is 0.86″. The corresponding static pressure rating required for a steel plate fan (*American Blower Co.*) is 0.86/1.36 or 0.64″ and for a Sirocco fan is 0.86/1.15 or .75″.



Full Housed Ventilating Fan

FIG. 49.

TABLE 22

SINGLE INLET AMERICAN SIROCCO FANS

Full Housed

(*American Blower Co.*)

No. of Fan	Wheel						Housing			Inlet				Outlet				
					Dia. of Shaft									Full Width				
	Diameter Inches	Width at Periphery (Full Width)	No. of Blades	Circumference In Feet	For Overhung Wheel	For All Other Arrangements	Over-all Height "A"	Width with Full Width Wheel "I"	Over-all Length B. + D.	Maximum Dia., Inches	Minimum Dia., Inches	Maximum Area, Square Feet	Minimum Area, Square Feet	Width and Height	Area Square Feet	Pulley Dia. and Face	To Center of Wheel "E"	Net Weight Lb. Wheel Full Width
4..	24	12	64	6.283	1 11/16	1 11/16	44¾	19	35 23/32	26⅜	21⅛	3.783	2.433	19 x 24	3.167	....	....	80
4½	27	13½	64	7.068	1 15/16	1 15/16	49 15/16	21⅜	40⅛	29¾	23¾	4.875	3.076	21⅜ x 27	4.008	....	....	120
5..	30	15	64	7.854	1 15/16	1 15/16	55⅛	23¾	44 9/16	33	26½	5.939	3.830	23¾ x 30	4.948	....	....	162
6..	36	18	64	9.425	2 3/16	2 3/16	65 9/16	28½	53 7/16	39⅝	31¾	8.563	5.498	28½ x 36	7.125	....	....	265
7..	42	21	64	10.995	2 7/16	2 3/16	75½	33½	62 9/32	46¼	37⅛	11.666	7.517	33¼ x 42	9.698	20 x 8	34½	340
8..	48	24	64	12.566	2 15/16	2 7/16	85 29/32	38	71 5/32	52¾	42⅜	15.176	9.793	38 x 48	12.666	22 x 9	39	480
9..	54	27	64	14.137	3 3/16	2 15/16	96 13/16	42¾	80	59⅜	47⅝	19.227	12.370	42¾ x 54	16.031	24 x 9	44	630
10..	60	30	64	15.708	3 7/16	3 3/16	107	47½	88⅞	66	52⅞	23.757	15.248	47½ x 60	19.791	26 x 10	48	830
11..	66	33	64	17.279	3 15/16	3 7/16	117 7/16	52¼	97¾	72⅝	58¼	28.766	18.506	52¼ x 66	23.947	28 x 10	53	1150
12..	72	36	64	18.850	4 7/16	3 15/16	127 27/32	57	106⅝	79¼	63½	34.254	21.992	57 x 72	28.500	30 x 11	57	1400
13..	78	39	64	20.420	4 15/16	4 7/16	138 1/32	61¾	115 15/32	85¾	68¾	40.103	25.779	61¾ x 78	33.448	32 x 11	61½	1800
14..	84	42	64	21.991	5 15/16	4 15/16	148 29/61	66½	124 5/16	92⅜	74¼	46.539	30.069	66½ x 84	38.791	34 x 12	66	2200
15..	90	45	64	23.562	5 15/16	4 15/16	158⅞	71¼	133 3/16	99	79⅜	53.454	34.363	71¼ x 90	44.531	36 x 12	70½	2600
16..	96	48	64	25.133	6 15/16	5 15/16	169 15/16	76	142 1/16	105⅝	84⅝	60.848	39.059	76 x 96	50.666	38 x 13	75⅞	3200
17..	102	51	64	26.704	6 7/16	6 7/16	180 11/32	80¾	150 29/32	112¼	90	68.720	44.178	80¾ x 102	57.198	40 x 13	80¼	3700
18..	108	54	64	28.274		6 15/16	190 21/32	85½	159 25/32	118¾	95¼	76.909	49.483	85½ x 108	64.125	42 x 14	84¾	4200
19..	114	57	64	29.845			200 15/16	90¼	168⅝	125⅜	100½	85.730	55.086	90¼ x 114	71.450	44 x 14	89¼	4900
20..	120	60	64	31.416			211¼	95	177½	132	105⅞	95.030	61.138	95 x 120	79.166	48 x 14	93½	5800

Dimensions are in inches, except where otherwise noted.
Letters refer to Fig. 49.

TABLE 23

SPEEDS, CAPACITIES AND HORSEPOWERS, SINGLE INLET STEEL PLATE FANS AT VARYING REVOLUTIONS

(*American Blower Co.*)

Air Temp. 70° F. Ratio total to static pressure = 1.36. Ratio velocity to static pressure = .36.
70 per cent opening

Fan Number	Dia. Wheel	Static Press.	½″	1″	1½″	2″
50	30	C. F. M.	3840	5425	6640	7650
		R. P. M.	471	665	816	945
		B. H. P.	88	2.48	4.55	7.00
60	36	C. F. M.	5475	7740	9460	10900
		R. P. M.	393	555	681	786
		B. H. P.	1.25	3.53	6.49	9.94
70	42	C. F. M.	7100	10020	12280	14150
		R. P. M.	336	475	583	675
		B. H. P.	1.62	4.58	8.35	12.93
80	48	C. F. M.	8640	12200	14950	17200
		R. P. M.	294	416	511	590
		B. H. P.	1.97	5.57	10.20	15.71
90	54	C. F. M.	11000	15540	19000	21900
		R. P. M.	262	370	454	525
		B. H. P.	2.52	7.08	13.00	20.00
100	60	C. F. M.	14050	19850	24300	28000
		R. P. M.	236	333	409	473
		B. H. P.	3 21	9.05	16.65	25.60
110	66	C. F. M.	16600	23500	28800	33100
		R. P. M.	214	303	371	430
		B. H. P.	3.80	10.75	19.70	30.25
120	72	C. F. M.	20300	28700	35100	40500
		R. P. M.	196	278	340	394
		B. H. P.	4.64	13.10	24.00	37.00
140	84	C. F. M.	27400	38700	47400	54500
		R. P. M.	168	238	292	337
		B. H. P.	6.25	17.75	32.40	49.80
160	96	C. F. M.	34500	48900	59800	68900
		R. P. M.	147	208	256	296
		B. H. P.	7.88	22.30	41.00	62.90
180	108	C. F. M.	42600	60300	73800	85000
		R. P. M.	131	185	227	262
		B. H. P.	9.75	27.55	50.50	77.60
200	120	C. F. M.	51600	73000	89400	103000
		R. P. M.	118	166	204	236
		B. H. P.	11.8	33.30	61.20	93.50
220	132	C. F. M.	61400	86800	106000	122200
		R. P. M.	107	151	185	214
		B. H. P.	14.0	39.60	72.50	111.50
240	144	C. F. M.	72000	101800	124500	143500
		R. P. M.	98	139	170	197
		B. H. P.	16.5	46.50	85.00	131.00

NOTE.—Any of the above fans, when running at the speed and pressure indicated, will deliver the volume of air and require no more power than given in the table.

Allowances must be made for the inefficiency of the motive power and for transmission losses between motive power and the fan.

This table is based on air volumes at 70° F. and is correct for altitudes up to and including 1000 feet.

The "per cent opening" refers to the opening used for the rating given from the test data.

TABLE 24

SPEEDS, CAPACITIES AND HORSEPOWERS, AMERICAN SIROCCO SINGLE INLET, STANDARD WIDTH FANS

(*American Blower Co.*)

Air temp. 70° F. Ratio total to static pressure = 1.15. Ratio velocity to static pressure = .15.
50 per cent opening

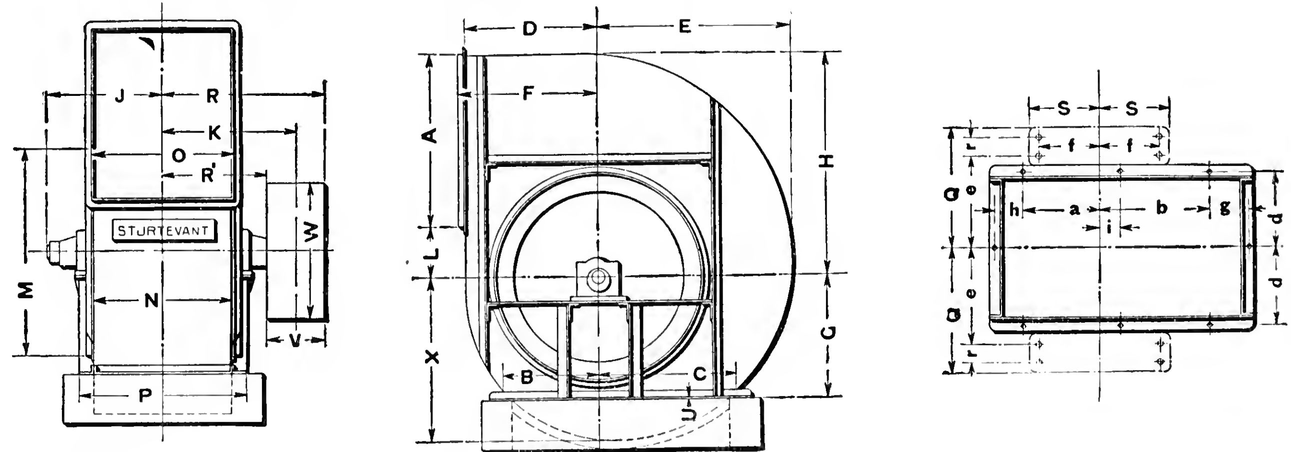
Fan No.	Dia. Wheel	Static Pressure Inches W. G.	¼	½	¾	1	1¼	1½	2
4.......	24	C. F. M.	2125	3480	4260	4920	5500	6020	6945
		R. P. M.	225	320	391	453	505	554	640
		B. H. P.	.244	.526	.97	1.485	2.08	2.725	4.19
4½.....	27	C. F. M.	3070	4410	5400	6205	6970	7620	8800
		R. P. M.	200	284	348	402	449	492	568
		B. H. P.	.308	.664	1.215	1.86	2.62	3.44	5.29
5.......	30	C. F. M.	3790	5450	6650	7690	8600	9416	10870
		R. P. M.	180	256	313	362	403	443	512
		B. H. P.	.378	.811	1.492	2.295	3.225	4.22	6.48
6.......	36	C. F. M.	5460	7835	9580	11060	12350	13540	15630
		R. P. M.	150	214	260	302	336	369	427
		B. H. P.	.541	1.06	2.14	3.28	4.6	6.01	9.27
7.......	42	C. F. M.	7425	10670	13050	15070	16800	18425	21260
		R. P. M.	129	183	223	259	288	316	366
		B. H. P.	.731	1.57	2.9	4.44	62.2	8.15	12.55
8.......	48	C. F. M.	9720	13920	17000	19700	22000	24100	27820
		R. P. M.	112	160	196	226	252	277	320
		B. H. P.	.954	2.045	3.76	5.78	8.14	10.61	16.32
9.......	54	C. F. M.	12250	17615	21500	24860	27800	30440	35140
		R. P. M.	100	142	174	201	224	246	285
		B. H. P.	1.206	2.58	4.73	7.26	10.22	13.38	20.5
10.......	60	C. F. M.	15150	21760	26500	30750	34300	37650	43400
		R. P. M.	90	128	156	181	202	222	256
		B. H. P.	1.473	3.17	5.81	8.95	12.55	16.48	25.25
11.......	66	C. F. M.	18350	26350	32200	37200	41500	45530	52550
		R. P. M.	82	117	142	165	184	202	233
		B. H. P.	1.78	3.82	7.05	10.78	15.15	19.85	30.45
12.......	72	C. F. M.	21800	31350	38300	44240	49400	54130	62500
		R. P. M.	75	107	130	151	168	185	214
		B. H. P.	2.11	4.54	8.35	12.81	18.0	23.55	36.2
13.......	78	C. F. M.	25600	36800	45000	52000	58100	63600	73500
		R. P. M.	69	99	120	140	155	171	197
		B. H. P.	2.465	5.31	9.8	15.0	21.1	27.6	42.5
14.......	84	C. F. M.	29650	42650	52100	60200	67300	73700	85000
		R. P. M.	64	92	112	130	144	158	183
		B. H. P.	2.87	6.15	11.33	17.4	24.45	32.0	49.2
15.......	90	C. F. M.	34100	49000	59900	69230	77500	84700	97800
		R. P. M.	60	86	104	121	135	148	171
		B. H. P.	3.27	7.06	13.0	19.93	28.0	36.6	56.4
16.......	96	C. F. M.	38800	55900	68100	78750	88400	96500	111800
		R. P. M.	56	80	98	112	126	139	160
		B. H. P.	3.73	8.0	14.8	22.65	31.8	41.6	65.1
17.......	102	C. F. M.	43800	63100	77000	89000	99900	109000	126000
		R. P. M.	53	75	92	106	119	130	150
		B. H. P.	4.21	9.03	16.75	25.6	35.9	46.9	73.5
18.......	108	C. F. M.	49200	70750	86400	99900	112000	122000	141500
		R. P. M.	50	71	87	100	112	123	142
		B. H. P.	4.72	10.15	18.8	28.65	40.3	52.6	82.5
19.......	114	C. F. M.	54750	78750	96200	111000	124500	136000	157200
		R. P. M.	47	67	83	95	106	117	134
		B. H. P.	5.25	11.3	20.9	31.9	45.0	58.6	91.8
20.......	120	C. F. M.	60600	87300	106800	123000	138000	150800	174500
		R. P. M.	45	64	78	91	101	111	128
		B. H. P.	5.82	12.5	23.1	35.4	49.7	65.0	101.8

NOTE.—Double inlet fans have approximately double the capacities of single inlet fans and require for their operation—under the same conditions—approximately twice the power.

The "per cent opening" refers to the opening used for the rating given from the test data.

STURTEVANT MULTIVANE FANS

SINGLE WIDTH, 7/8 HOUSING, OVERHUNG PULLEY DOUBLE INLET, TOP HORIZONTAL DISCHARGE



NOTE — R' = END OF BEARING TO ℄ FAN
SIZES 18 AND ABOVE HAVE BEARING SUPPORTS OF HEAVIER CONSTRUCTION

TABLE 25

ALL DIMENSIONS ARE IN INCHES

Fan Size	Extreme Wheel Dia.	Shaft Dia. Bearing	Shaft Dia. Wheel	A	B	C	D	E	F	G	H	J	K
10	46 1/2	2 11/16	2 11/16	40	22 1/4	32 1/2	30 1/2	45	32 1/2	29 1/8	52 1/4	27 1/8	29
11	52 1/2	2 15/16	2 15/16	45	25 1/4	36 3/4	34 3/8	50 5/8	36 3/8	32 1/8	58 3/4	30	32
12	58 1/4	2 15/16	2 15/16	49 7/8	27 1/2	40 1/2	38 1/8	56 1/8	40 1/8	36 5/8	65 1/8	32	34
13	64	3 7/16	3 7/16	54 3/4	30 1/2	41 1/2	42	61 3/4	44 1/2	39 7/8	71 5/8	35 1/4	37 3/4
14	69 3/4	3 7/16	3 7/16	59 5/8	33 1/4	48 3/4	45 3/4	67 1/2	48 1/4	42 7/8	78 3/8	37 1/4	39 3/4
15	75 3/4	3 15/16	3 15/16	64 1/2	36 1/2	53 1/2	49 5/8	73 1/8	52 1/8	45 5/8	84 7/8	41 3/8	43 3/4
16	81 1/2	4 1/2	4 1/2	69 1/2	39	57 1/4	53 3/8	78 5/8	55 7/8	48 7/8	91 1/4	44 5/8	46 3/4
17	87 1/2	5	5	74 1/2	42	61 1/2	57 1/8	84 1/8	60 5/8	52 1/8	97 5/8	48 1/4	50 1/4
18	93 1/4	5 1/2	5 1/2	79 1/4	44 3/4	65 5/8	60 7/8	89 7/8	64 3/8	55 1/4	104 3/4	58 7/8	....
20	105	6	6	89 1/8	50 3/4	74 1/4	68 1/2	101	72	61 1/4	117 1/4	63 1/8	....
22	116 1/2	6	6 1/2	99 1/8	56 1/4	82 1/4	76 1/4	112 1/4	79 3/4	68	130 1/4	67 1/8	....
24	128	6	7 1/2	108 3/4	62	90 3/4	83 3/4	123 1/2	88 1/4	74	143 3/8	71 1/8	....
26	140	6	8	118 1/2	67 3/4	99 1/4	91 3/8	134 5/8	95 7/8	80	156 1/4	75 1/8	....

Fan Size	L	M	N	O	P	Q	R	R'	S	U	V	W	X
10	12 1/2	48 1/2	32 1/4	32 3/4	39 1/4	....	34 1/4	....	....	5/16	10 1/2	20	37 3/4
11	14	54 1/2	36 1/4	36 3/4	43 1/4	....	38	....	....	5/16	12	22	42 1/2
12	15 1/2	60 1/2	40 1/4	40 3/4	50 1/4	....	40	....	....	3/8	12	24	47 1/8
13	17 1/8	66 1/2	44 1/4	44 3/4	54 1/4	....	44 1/2	....	....	3/8	13 1/2	28	51 7/8
14	19	72 1/2	48 1/4	48 3/4	58 1/4	....	46 1/2	....	....	3/8	13 1/2	32	56 5/8
15	20 5/8	78 1/2	52 1/4	52 3/4	64 1/4	....	51	....	....	7/16	14 1/2	36	61 3/8
16	22	84 3/4	56 1/4	56 3/4	68 1/4	....	54	....	....	7/16	14 1/2	42	66
17	23 7/16	90 3/4	60 1/4	60 7/8	72 1/4	....	57 1/2	....	....	7/16	14 1/2	48	70 5/8
18	25 7/16	96 7/8	64 1/4	64 7/8	76 1/4	55 7/8	....	54 5/8	26 1/2	7/16	....	..	75 3/8
20	28 7/16	109	72 1/4	72 7/8	84 1/4	60 7/8	....	58 5/8	28 1/4	7/16	....	..	84 3/4
22	31 11/16	121 1/8	80 1/4	80 7/8	92 1/4	65 7/8	....	62 5/8	30	7/16	....	..	94 1/4
24	35	133 1/8	88 1/4	89	100 1/4	70 7/8	....	66 5/8	32 1/8	7/16	....	..	103 5/8
26	38 1/8	145 1/4	96 1/4	97	108 1/4	75 7/8	....	70 5/8	33 3/4	7/16	....	..	113

Fan Size	a	b	d	e	f	g	h	i	r	Anchor Bolts Ped'st'l	Anchor Bolts Fan
10	19	19	18 5/8	....	....	15 3/4	5 3/4	....	....	....	7/8
11	20 1/4	21	20 5/8	....	....	18 1/4	7 1/2	....	....	....	7/8
12	21 1/2	24	22 7/8	....	....	19 1/4	8 3/4	....	....	....	7/8
13	23	26	24 7/8	....	....	21 1/4	10 1/4	....	....	....	7/8
14	23 1/2	28	26 7/8	....	....	23 1/2	12 1/2	....	....	....	7/8
15	25	30 7/8	29 5/8	....	....	26 1/8	15	....	....	....	7/8
16	27	33 7/8	31 5/8	....	....	26 7/8	15 1/2	....	....	....	7/8
17	28 1/2	35	33 5/8	....	....	30	17	....	....	....	1
18	33	46	35 5/8	40 7/8	24 1/2	23 1/8	15 1/4	6 1/2	12	1 1/4	1
20	36	52	39 5/8	44 7/8	26 1/4	25 3/4	18 1/4	8	13	1 1/4	1
22	40	58	43 5/8	48 7/8	28	27 3/8	19 3/4	9	14	1 1/4	1
24	44	64	47 5/8	52 7/8	30 1/8	30 1/4	21 1/2	10	14	1 1/4	1 1/8
26	46	70	51 5/8	56 7/8	31 3/4	32 3/4	25 1/4	12	15	1 1/4	1 1/8

Data and dimensions for *B. F. Sturtevant Co.'s* "Multivane" Fans are given in Table 25, preceding.

TABLE 26

INSTALLATION DATA

1-Inch Static Pressure (Water Gage), Air at 65° F.

Maximum resistance for public building heating systems with air washers. Minimum resistance for factory heating systems. Ordinary resistance for forced draft with low grade run of mine bituminous coal.

Size Fan	R. P. M.	Vol. Cu. Ft. per Min.	Brake H. P.	Size Motor	Size Vertical Engine			Size Horizontal Engine			Size Pulley	
					80 Lb. Pressure	100 Lb. Pressure	120 Lb. Pressure	80 Lb. Pressure	100 Lb. Pres.	120 Lb. Pres.	Dia. In.	Face In.
2	1007	1025	.33	B	....	....	....	....	....	....	6	3½
	1047	1235	.42	B	....	....	....	....	....	....	6	3½
	1115	1510	.58	B	....	....	....	....	....	....	6	3½
	1230	1870	.90	B	....	....	....	....	....	....	6	3½
3	804	1600	.52	B	3 x 2½	3 x 2½	3 x 2½	....	....	....	7	4
	836	1930	.65	B	3 x 2½	3 x 2½	3 x 2½	....	....	....	7	4
	892	2360	.90	B	3 x 2½	3 x 2½	3 x 2½	....	....	....	7	4
	984	2920	1.4	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	7	4
4	668	2320	.75	B	3 x 2½	3 x 2½	3 x 2½	....	....	....	8	5
	695	2790	.95	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	8	5
	741	3420	1.3	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	8	5
	818	4230	2.0	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	8	5
5	573	3130	1.0	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	10	5
	597	3770	1.3	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	10	5
	636	4670	1.8	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	10	5
	702	5710	2.8	H	3 x 2½	3 x 2½	3 x 2½	....	....	....	10	5
6	502	4110	1.3	H	3½ x 3	3 x 2½	3 x 2½	....	....	....	12	6
	522	4950	1.7	H	3½ x 3	3 x 2½	3 x 2½	....	....	....	12	6
	557	6050	2.3	H	3½ x 3	3½ x 3	3 x 2½	....	....	....	12	6
	615	7500	3.6	H	4 x 4	4 x 4	3½ x 3	....	....	....	12	6
6½	448	5250	1.7	H	3½ x 3	3½ x 3	3 x 2½	....	....	....	13	6½
	466	6320	2.1	H	4 x 4	3½ x 3	3 x 2½	....	....	....	13	6½
	497	6740	3.0	H	4 x 4	4 x 4	3½ x 3	....	....	....	13	6½
	549	9570	4.6	H	......	4 x 4	......	....	....	....	13	6½
7	402	6410	2.1	H	4 x 4	3½ x 3	3½ x 3	....	....	....	14	7½
	418	7720	2.6	H	4 x 4	4 x 4	3½ x 3	....	....	....	14	7½
	445	9460	3.6	H	5 x 5	4 x 4	4 x 4	....	....	....	14	7½
	492	11800	5.6	H	5 x 5	5 x 5	4 x 4	....	....	....	14	7½
8	335	9300	3.0	C	5 x 5	4 x 4	4 x 4	....	....	....	16	9
	348	11200	3.8	C	5 x 5	5 x 5	4 x 4	....	....	....	16	9
	371	13700	5.2	C	5 x 5	5 x 5	5 x 5	....	....	....	16	9
	410	16950	8.2	C	6 x 5	5 x 5	5 x 5	....	....	....	16	9
9	287	12580	4.1	C	5 x 5	5 x 5	5 x 5	....	....	....	18	10½
	298	15150	5.1	C	6 x 5	5 x 5	5 x 5	....	....	....	18	10½
	318	18550	7.1	C	6 x 5	6 x 5	5 x 5	....	....	....	18	10½
	352	22950	11.1	C	7 x 6	7 x 5	6 x 5	....	....	....	18	10½
10	251	16500	5.4	C	6 x 5	6 x 5	5 x 5	....	....	....	20	10½
	261	19880	6.7	C	7 x 5	6 x 5	5 x 5	....	....	....	20	10½
	278	24320	9.3	C	7 x 6	7 x 5	6 x 5	....	....	....	20	10½
	308	30100	14.5	C	8 x 6	7 x 6	6 x 6	....	....	....	20	10½
11	223	20750	6.7	C	7 x 6	6 x 5	6 x 5	....	....	....	22	12
	231	25000	8.5	C	7 x 6	7 x 5	6 x 5	....	....	....	22	12
	247	30600	11.7	C	8 x 6	7 x 6	6 x 6	....	....	....	22	12
	272	37850	18.2	C	9 x 7	8 x 7	7 x 7	8 x 10	....	....	22	12

Effect of Temperature on Fan Performance. The tables of fan manufacturers are based on a temperature of 70° F. and a barometric pressure of approximately 29.9″ Hg.

Fans used for handling air in heating systems and particularly for induced draft installations deal with air at considerably higher temperatures.

For *a constant pressure* the tabulated speed, volume, and horsepower must be multiplied by the square root of the ratio of the density of air at the rated temperature to the density at the desired temperature. As the density varies inversely as the absolute temperature the following relation results:

$$\sqrt{\frac{\text{density at rated temp. } t_1}{\text{density at required temp. } t_2}} = \sqrt{\frac{460 + t_2}{460 + t_1}}$$

The following table gives the value of the above expression based on a tabulated temperature $t = 70°$ F.:

TABLE 27

FIGURED FOR DRY AIR AT SEA LEVEL. BAR. 29.92″

Temperature	$\sqrt{\text{Ratio}}$	Temperature	$\sqrt{\text{Ratio}}$	Temperature	$\sqrt{\text{Ratio}}$
−20	.911	130	1.055	400	1.273
−10	.921	140	1.064	420	1.288
0	.932	150	1.073	440	1.303
10	.942	160	1.082	460	1.317
20	.952	170	1.090	480	1.331
30	.961	180	1.098	500	1.345
40	.971	190	1.106	520	1.359
50	.981	200	1.114	540	1.373
60	.990	260	1.165	560	1.387
70	1.000	280	1.181	580	1.400
80	1.009	300	1.197	600	1.413
90	1.018	320	1.213	620	1.426
100	1.028	340	1.228	640	1.439
110	1.037	360	1.243	660	1.452
120	1.046	380	1.258	680	1.465
				700	1.478

TABLE 28

TABLE SHOWING VOLUME, DENSITY AND BAROMETRIC PRESSURE OF AIR AT VARIOUS ALTITUDES WITH CORRESPONDING VELOCITY CONSTANTS, TEMPERATURE REMAINING CONSTANT AT APPROXIMATELY 70° FAHRENHEIT

Altitude Above Sea Level	Barometer, Inches Hg.	Weight per Cu. Ft.	Volume, Cu. Ft. per Pound	Velocity Constant	Speed Ratio
0	30.00	0.07500	13.33	4006	1.000
500	29.44	.07360	13.58	4040	1.010
1000	28.87	.07218	13.85	4080	1.020
1500	28.33	.07083	14.12	4120	1.030
2000	27.80	.06950	14.39	4165	1.040
2500	27.27	.06818	14.67	4200	1.048
3000	26.75	.06688	14.95	4240	1.058
3500	26.23	.06558	15.25	4280	1.068
4000	25.74	.06435	15.54	4320	1.078
4500	25.26	.06315	15.84	4360	1.084
5000	24.77	.06193	16.14	4410	1.100
5500	24.31	.06080	16.44	4450	1.110
6000	23.85	.05963	16.76	4500	1.122
6500	23.39	.05848	17.10	4540	1.133
7000	22.94	.05735	17.44	4580	1.145
7500	22.52	.05630	17.76	4630	1.155
8000	22.08	.05520	18.12	4670	1.165
8500	21.67	.05418	18.46	4710	1.175
9000	21.27	.05318	18.80	4760	1.186
9500	20.87	.05218	19.16	4800	1.197
100000	20.45	.05113	19.56	4850	1.210

Example. Assume 20,800 cu. ft. of air required at 550° F. and 1″ pressure. From Table 27 the ratio between the speeds necessary to produce the same pressure for air at 550° and air at 70° F. is 1.38. $\frac{20{,}800}{1.38} = 15{,}072$ cu. ft. at 70° F. The fan selected for this work from Table 24 is a No. 7 which will

handle 15,070 cu. ft. at 70° F. and 1″ water pressure requiring a speed of 259 r. p. m. and 4.44 brake horsepower.

The *required speed at* 550° F. i 259 × 1.39 or 357 and *the required brake horsepower is* 4.44 × 1.38 or 6.13.

The values in column headed "Speed Ratio" (Table 28) represent the ratio of speeds necessary for a given pressure at various altitudes, taking the speed at sea level as unity.

These ratios can be used in connection with those given in Table 27 in problems involving high altitudes and temperature. For instance, the speed ratio for 7000 ft. altitude and 500° Fahr. = 1.145 × 1.345.

THE DESIGN OF AIR DUCTS

There are *two schemes* used in the proportioning of air ducts, based on (1) velocity and (2) friction pressure loss.

Velocity Method. One method is to arbitrarily fix the velocity in the various sections reducing the velocity from the point of leaving the fan to the point of discharge to the room. See Table 19 for velocities. In this case the pressure loss of each section is calculated separately and the total loss found by adding together the losses of the various sections.

For a "single duct system" such as shown by Fig. 50 the method employed is to proportion the longest duct, for certain velocities first, and determine the total pressure required at the plenum chamber, then the remaining ducts are so proportioned that their total pressure is the same as calculated for the longest duct. It is here assumed that the longest duct will have the greatest friction pressure loss. This may not be true as in the case of a shorter duct containing several sharp turns in which event all ducts should be proportioned for the total pressure, found necessary, in this duct. If the velocities through the free area of the register grills are all to be the same, which is customary practice, and all ducts are proportioned for the same velocities the friction loss will naturally be less in the remaining ducts and an added resistance is necessary to increase the total pressure to the amount determined upon. This is accomplished by dampering. In order to obviate the necessity of dampering and at the same time economize in the size of ducts the following method may be used:

The vertical flues or risers are all figured for the same velocity as before, as well as the velocity through the free area of register grill, and the extra or added resistance necessary in the shorter ducts provided for by increasing the velocities in the horizontal connections to the base of the risers.

Friction Pressure Loss Method. The other method refers particularly to a "trunk line" system of distribution in which case the duct may be proportioned for *equal friction pressure loss per foot of length.*

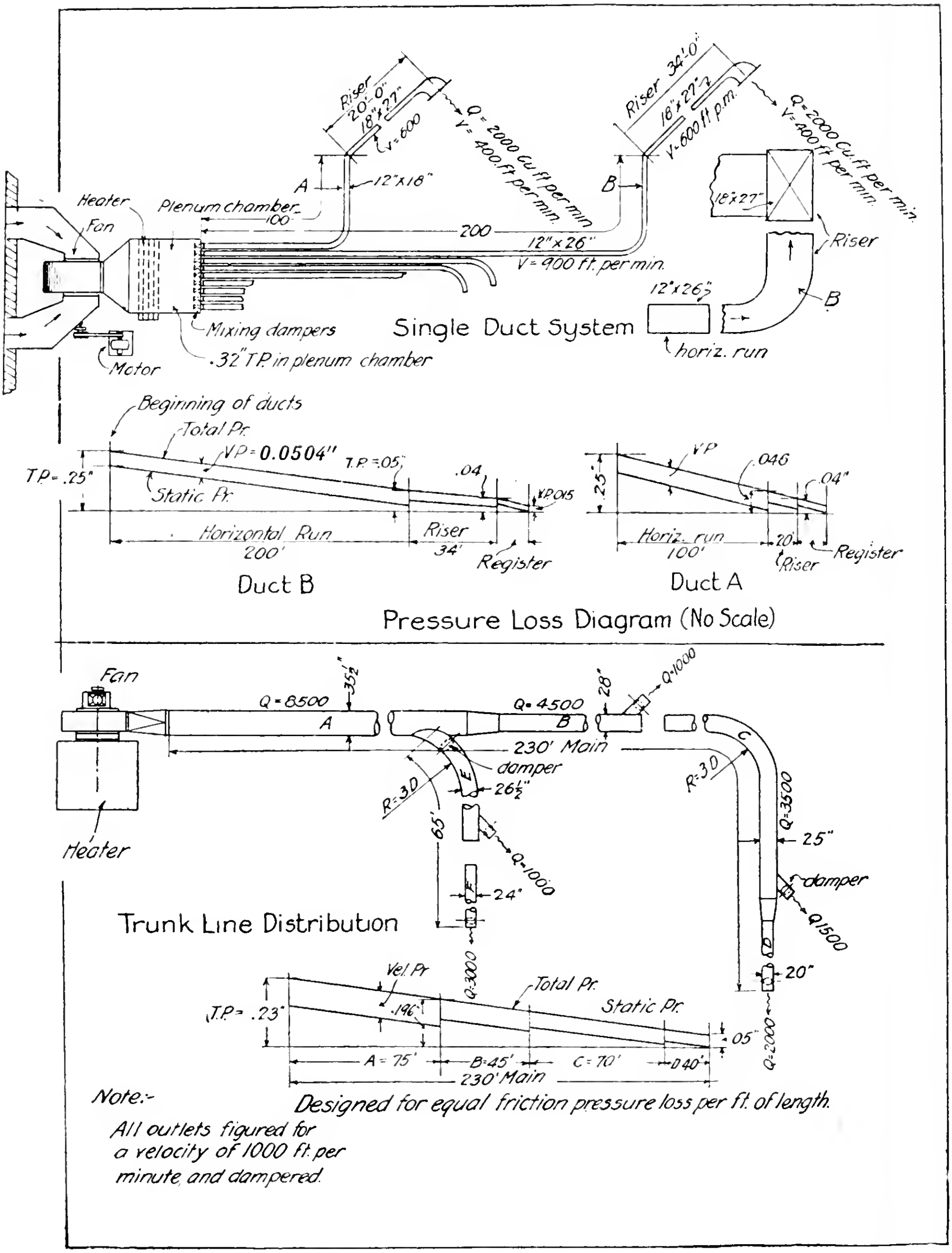
This method is a convenient one in proportioning mains and branches and in which it is unnecessary to calculate the friction loss of each section independently. The reduction in velocity as the pipe sizes become smaller is automatically accomplished by this method.

The velocity in the last or most remote section from the fan is fixed upon and the size and friction pressure loss per 100 ft. determined. Knowing the quantities of air to be transmitted by the various sections it is only necessary then to read from the chart (Fig. 38), the various diameters necessary to give the same friction pressure loss per 100 ft. as determined for the last section.

The total loss in the duct alone will then be the total length, *which includes any allowance for ells or turns,* divided by 100 times the pressure loss per 100 ft. as read from the chart for the last section.

The two examples following will serve to make clear the methods, as outlined, for the proportioning of air ducts

Example. (Velocity as a basis for the design.) Refer to Fig. 50 for a single duct system. The risers will be figured for a velocity of 600 ft. per min. and 400 ft. per min. through the free area of reg-



FIGS. 50 AND 51.

ister grill, and the longest main "B" for 900 ft. per min. As this duct is to transmit 2000 cu. ft. per min. the required area of riser is 2000/600 or 3.33 sq. ft.; an 18" x 27" duct gives practically this area. The required area of the horizontal run is 2.22 sq. ft. size of duct 12" x 26 ".

The pressure loss in the riser may be determined by means of the chart (Fig. 38), as follows: Locate the intersection of the 18" "height of duct line" and the 27" "width of duct line," then pass vertically downward to the "$\sqrt{\frac{A^3}{R}}$" line and parallel with the width of duct line to the intersection with the horizontal 2000 cu. ft. per min. line, then vertically downward and read 0.02" as the pressure loss per 100 ft. for a smooth duct, adding 50 per cent as an allowance for roughness of joints, etc., gives 0.03"; the measured length of the riser is 27.5 + 5 × 1.5 (height of duct) as an allowance for turn at the top, gives 34. ft. as the equivalent length on which to figure the friction loss. The estimated pressure loss in the riser will be 34 / 100 × 0.03 or 0.01".

The loss in the register grill (Fig. 39), is approximately 1.5 × the velocity pressure corresponding to a velocity of 400 ft. per min. or 1.5 × 0.015 or 0.023". The total pressure at the end of riser, or in register box is 0.015 + 0.023 = 0.038 say 0.04". The loss in the horizontal run as read from the chart for a 12" x 26" duct is 0.063" per 100 ft. and with an addition of 50 per cent for roughness gives 0.10" (nearly). The length of the horizontal run *including the allowance for ells*, is 200 ft. The loss is therefore 2 × 0.10 or 0.20". The total pressure required at the entrance to ducts is therefore *the sum of all the losses plus the final velocity pressure* or .023 + 0.01 + 0.20 + 0.015 = 0.25". The total pressure in the plenum chamber will be somewhat greater when the loss at entrance to ducts is taken into account.

Having established the total pressure required at the beginning of the ducts the size of any shorter ducts, as "A," may be obtained by using the same velocities and dampering them to give the added resistance necessary to bring up the total pressure to 0.25" or allowing the same velocity or 600 ft. per min. for the riser and designing the horizontal run to provide the necessary friction to increase the total pressure to the amount stated. Using the latter scheme we have for the loss in the riser "A," 0.03 × 20 / 100 or 0.006". The total head at the base of the riser is 0.04 + 0.006 or 0.046". The pressure loss in the horizontal run must therefore equal 0.25 − 0.046 or 0.204". The length of the horizontal run is 100 ft. including allowance for turns or ells.

The loss per 100 ft. must therefore be 0.204" and as this includes the 50 per cent allowance for rough joints, etc., the corresponding pressure loss on the chart will be 0.204 / 1.50 or 0.14" (nearly). Enter the chart Fig. 38 at this pressure loss and pass vertically upward to the intersection with the 2000 cu. ft. per min. line, then follow to the left parallel with the "diameter or width of duct line" to the intersection with the $\sqrt{\frac{A^3}{R}}$ line, then vertically to the intersection with the "12" height of duct line" and read 18" as the required width, the velocity being 2000/(1. × 1.5) or 1333 ft. per min. The actual total pressure in the plenum chamber must be higher than 0.25 to allow for the loss by entry to the ducts, which should be figured on a basis of the highest duct velocity or 1333. ft. per min. The velocity pressure is 0.11" and the loss by entry will be approximately 0.47 × 0.11 or 0.052". The dampers reduce the free area somewhat and increase the friction loss. Allowing for this by making the entry loss 0.07 gives 0.25 + 0.07 or 0.32" for the total pressure required in the plenum chamber. The loss through a 4 section pipe coil heater at 1200 ft. per min. velocity is 0.30" water. The loss in connection between fan and heater is about 0.48 × the velocity pressure of air leaving fan outlet. The velocity of the air leaving outlet is approximately 2000 ft. per min.; the corresponding velocity pressure is .25" (approx.) This loss is then 0.48 × 0.25 or 0.12". The loss in the connections to the suction side of fan for tne layout shown will be assumed the same or 0.12". The total estimated fan pressure required is therefore 0.32 + 0.30 + 0.12 + 0.12 = 0.86". The corresponding static pressure rating of the fan required, if a Sirocco fan is to be used, is 0.86 / 1.15 or ¾".

Example. (Friction Pressure Loss as a basis for the design.) Let it be required to proportion the duct sizes for the layout shown (Fig. 51), assuming a final velocity at the last outlet section "D" of approx. 1000 ft. per min. The various quantities of air "Q" cu. ft. per min. for the various sections are given on the drawing. The area of the last section will be 2000 / 1000 or 2 sq. ft. corresponding to a diameter of 20". The actual area is 2.18 sq. ft. and the corresponding velocity is 2000 / 2.18 or 917 ft. per min., the corresponding velocity pressure is 0.05" at 70° F. Entering the chart (Fig. 38) at the left at 2000 cu. ft. pass horizontally to the left to the intersection with the 20" diameter line and

read 0.048″ as the loss per 100 ft. in a smooth sheet steel duct. The design is to be based on equal friction pressure loss per ft. of length therefore all readings for duct sizes will be made from the 0.048″ line. The various sizes and length of sections, including the allowance for ells, for the main is given by the following table:

TABLE 29

TRUNK LINE SYSTEM (FIG. 51)

Section	Quantity of Air, Cubic Feet per Minute	Diameter, Inches	Velocity, Feet per Minute	Measured Length + Allowance for Ells
A	8500	35½	1250	75′
B	4500	28	....	45′
C	3500	25	....	70′ + (5 x 2)
D	2000	20	917	40′

Total length = 240′.

The total loss of pressure is 240 / 100 × 0.048 or 0.12″ plus 50 per cent allowance for roughness of pipe and reducers gives 0.18″.

The total pressure at the beginning of the main is therefore 0.18 + 0.05 or 0.23″ say ¼″. The branch main *E-F* is proportioned by first ascertaining the total pressure existing in the main at the point where the branch is taken off. The total pressure at this point will equal the sum of the loss by entry to branch plus the loss in the branch plus the final velocity pressure. Using the same final velocity as before (approx. 1000 ft. per min.) the velocity pressure is 0.05″.

The size of the last section "*F*" of the branch is 3000/1000 = 3 sq. ft. area or 24″ diam.

The total pressure in the main at point where branch is taken off is equal to total pressure at the beginning of the main minus the pressure loss of the first section *A*, 75 feet in length, or 0.23 − (75/100 × 1.5 × 0.048) or 0.176″. An allowance of 50 per cent is made for friction.

The loss by entry to a 45° branch, Fig. 39, is 0.22 × 0.05 or 0.011″. The loss of pressure in the turn, if made with a radius $R = 3D$ will be approximately 0.15 × 0.05″ *V. P.* or 0.008″.

The loss in section "*F*" from chart, is 25 / 100 × 0.048 or 0.012″ + 50 per cent or 0.018″.

The loss in section "*E*" must therefore be equal to 0.176″ − [0.011 + 0.008 + 0.018 + .05 (Final *V. P.*)] or 0.089″. This section handles 4000 cu. ft. per min. and is 40 ft. long. The loss per 100 ft. must therefore be equal to 100/40 × 0.087 or 0.22″, and, as this includes the 50 per cent allowance the corresponding pressure loss line on the chart to use is 0.22″/1.50 or 0.147″.

The required diameter of section "*E*" is therefore 21″ by this method.

This method is not considered practical. The branch would be designed for equal friction loss per 100 ft. based on the diameter of the last section *F* of the branch or 24″. This gives 26½″ for the diameter of section *E*. A damper placed in section *E* gives the resistance required to bring up the total pressure to 0.176″ and reduce the flow to the required amount or 4000 cu. ft. per min.

Sheet Metal Pipes and Ducts. The recommended gage (U. S. sheet metal gage) for various sizes of galvanized sheet steel pipes for heating and ventilating work, blowpipe and exhaust work is given by the table following: (*American Blower Co.*)

TABLE 30

METAL GAGES FOR DUCTS

HEATING AND VENTILATING		BOTH	BLOW PIPING AND EXHAUST WORK	
Diameter Inches	U. S. S. Gage Number	Thickness Inches and Weight per Square Feet	Diameter Inches	U. S. S. Gage Number
6–18	26	0.0187″– .91 lb.	3– 5	26
19–36	24	.025″ –1.16 "	6– 8	24
38–48	22	.0312″–1.41 "	9–15	22
50–60	20	.0375″–1.66 "	16–24	20
63–72	18	.05″ –2.16 "	26–30	18

TABLE 31

AREA AND CIRCUMFERENCE OF CIRCLES

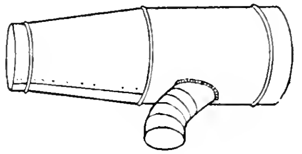
Diameter		Area		Circumference		
Feet	Ins.	Square Inches	Square Feet	Inches	Feet and Dec.	Feet and Inches
.0833	1	.7854	.0054	3.1416	.2618	0' 3 1/8"
.1667	2	3.14	.0218	6.28	.5236	0' 6 7/8"
.2500	3	7.07	.0491	9.42	.7854	0' 9 7/16"
.3333	4	12.57	.0873	12.57	1.047	1' 0 9/16"
.4167	5	19.63	.1364	15.71	1.309	1' 3 1/16"
.5000	6	28.27	.1964	18.85	1.571	1' 6 7/8"
.5833	7	38.48	.2673	21.99	1.833	1' 10"
.6667	8	50.27	.3491	25.13	2.094	2' 1 1/8"
.7500	9	63.62	.4418	28.27	2.356	2' 4 1/4"
.8333	10	78.54	.5454	31.42	2.618	2' 7 7/16"
.9167	11	95.03	.6600	34.56	2.880	2' 10 9/16"
1.000	12	113.1	.7854	37.70	3.142	3' 1 5/8"
1' 1"	13	132.7	.9218	40.84	3.403	3' 4 5/8"
1' 2"	14	153.9	1.069	43.98	3.665	3' 8"
1' 3"	15	176.7	1.227	47.12	3.927	3' 11"
1' 4"	16	201.0	1.396	50.26	4.189	4' 2 1/8"
1' 5"	17	226.9	1.576	53.41	4.451	4' 5 3/8"
1' 6"	18	254.4	1.767	56.55	4.712	4' 8 1/2"
1' 7"	19	283.5	1.969	59.69	4.974	4' 11 5/8"
1' 8"	20	314.1	2.182	62.83	5.236	5' 2 3/8"
1' 9"	21	346.3	2.405	65.97	5.498	5' 5 7/8"
1' 10"	22	380.1	2.640	69.11	5.760	5' 9"
1' 11"	23	415.4	2.885	72.26	6.021	6' 2 1/4"
2' 0"	24	452.3	3.142	75.40	6.283	6' 3 3/8"
2' 1"	25	490.8	3.409	78.54	6.545	6' 6 1/2"
2' 2"	26	530.9	3.687	81.68	6.807	6' 9 3/8"
2' 3"	27	572.5	3.976	84.82	7.069	7' 0 3/4"
2' 4"	28	615.7	4.276	87.96	7.330	7' 3 7/8"
2' 5"	29	660.5	4.587	91.11	7.592	7' 7"
2' 6"	30	706.8	4.909	94.25	7.854	7' 10 1/4"
2' 7"	31	754.7	5.241	97.39	8.116	8' 1 3/8"
2' 8"	32	804.2	5.585	100.5	8.376	8' 4 1/2"
2' 9"	33	855.3	5.940	103.7	8.639	8' 7 3/8"
2' 10"	34	907.9	6.305	106.8	8.901	8' 10 3/4"
2' 11"	35	962.1	6.611	109.9	9.163	9' 1 7/8"
3' 0"	36	1017.8	7.069	113.1	9.425	9' 5"
3' 1"	37	1075.2	7.467	116.2	9.686	9' 8 1/4"
3' 2"	38	1134.1	7.876	119.4	9.948	9' 11 3/4"
3' 3"	39	1194.5	8.296	122.5	10.21	10' 2 1/2"
3' 4"	40	1256.6	8.727	125.7	10.47	10' 5 5/8"
3' 5"	41	1320.2	9.168	128.8	10.73	10' 8 3/8"
3' 6"	42	1385.4	9.621	131.9	10.99	10' 11 7/8"
3' 7"	43	1452.2	10.08	135.1	11.26	11' 3"
3' 8"	44	1520.5	10.56	138.2	11.52	11' 6 1/8"
3' 9"	45	1590.4	11.04	141.4	11.78	11' 9 3/8"
3' 10"	46	1661.9	11.54	144.5	12.04	12' 0 1/2"
3' 11"	47	1734.9	12.05	147.6	12.30	12' 3 5/8"
4' 0"	48	1809.5	12.57	150.8	12.57	12' 6 3/4"
4' 1"	49	1885.7	13.10	153.9	12.83	12' 9 7/8"
4' 2"	50	1963.5	13.64	157.1	13.09	13' 1"

Diameter		Area		Circumference		
Feet	Ins.	Sq. Ins.	Sq. Feet	Ins.	Feet and Dec.	Feet and Inches
4' 3"	51	2042	14.19	160.2	13.35	13' 4 1/8"
4' 4"	52	2124	14.75	163.4	13.61	13' 7 1/4"
4' 5"	53	2206	15.32	166.5	13.88	13' 10 1/2"
4' 6"	54	2290	15.90	169.6	14.14	14' 1 5/8"
4' 7"	55	2376	16.50	172.8	14.40	14' 4 5/8"
4' 8"	56	2463	17.10	175.9	14.66	14' 7 7/8"
4' 9"	57	2552	17.72	179.0	14.92	14' 11"
4' 10"	58	2642	18.35	182.2	15.18	15' 2 1/8"
4' 11"	59	2734	18.99	185.3	15.45	15' 5 1/4"
5' 0"	60	2827	19.63	188.5	15.71	15' 8 1/8"
5' 1"	61	2922	20.29	191.6	15.97	15' 11 3/8"
5' 2"	62	3019	20.97	194.8	16.33	16' 2 3/4"
5' 3"	63	3117	21.65	197.9	16.49	16' 5 3/4"
5' 4"	64	3217	22.34	201.1	16.76	16' 9"
5' 5"	65	3318	23.04	204.2	17.02	17' 0 1/8"
5' 6"	66	3421	23.76	207.3	17.28	17' 3 1/4"
5' 7"	67	3526	24.48	210.5	17.54	17' 6 3/8"
5' 8"	68	3632	25.22	213.6	17.80	17' 9 5/8"
5' 9"	69	3739	25.97	216.8	18.06	18' 0 3/4"
5' 10"	70	3848	26.73	219.9	18.33	18' 3 7/8"
5' 11"	71	3959	27.49	223.1	18.59	18' 7 1/8"
6' 0"	72	4072	28.27	226.2	18.85	18' 10 1/8"
6' 1"	73	4185	29.07	229.3	19.11	19' 1 1/4"
6' 2"	74	4301	29.87	232.5	19.37	19' 4 3/8"
6' 3"	75	4418	30.68	235.6	19.63	19' 7 1/2"
6' 4"	76	4536	31.50	238.8	19.90	19' 10 5/8"
6' 5"	77	4657	32.34	241.9	20.16	20' 1 7/8"
6' 6"	78	4778	33.18	245.0	20.42	20' 4 7/8"
6' 7"	79	4902	34.04	248.2	20.68	20' 8 1/8"
6' 8"	80	5027	34.91	251.3	20.94	20' 11 1/2"
6' 9"	81	5158	35.78	251.5	21.21	21' 2 3/8"
6' 10"	82	5281	36.67	257.6	21.47	21' 5 1/2"
6' 11"	83	5411	37.57	260.8	21.73	21' 8 3/4"
7' 0"	84	5542	38.48	263.9	21.99	21' 11 7/8"
7' 1"	85	5674	39.41	267.0	22.25	22' 3"
7' 2"	86	5809	40.34	270.2	22.51	22' 6 1/8"
7' 3"	87	5945	41.28	273.3	22.71	22' 9 1/4"
7' 4"	88	6082	42.24	276.5	23.04	23' 0 3/8"
7' 5"	89	6221	43.20	279.1	23.30	23' 2 1/8"
7' 6"	90	6362	44.18	282.7	23.56	23' 6 3/4"
7' 7"	91	6504	45.17	285.9	23.82	23' 9 7/8"
7' 8"	92	6648	46.16	289.0	24.09	24' 1 1/8"
7' 9"	93	6798	47.17	292.2	24.35	24' 4 1/8"
7' 10"	94	6940	48.19	295.3	24.61	24' 7 1/4"
7' 11"	95	7088	49.22	298.5	24.87	24' 10 3/8"
8' 0"	96	7238	50.27	301.6	25.13	25' 1 1/2"
8' 1"	97	7390	51.32	304.7	25.39	25' 4 5/8"
8' 2"	98	7543	52.38	307.9	25.66	25' 7 7/8"
8' 3"	99	7698	53.46	311.0	25.92	25' 11"
8' 4"	100	7855	54.54	314.2	26.18	26' 2 1/8"

Diameter		Area		Circumference		
Feet	Ins.	Sq. Inches	Sq. Feet	Ins.	Feet and Dec.	Feet and Inches
8' 5"	101	8011.9	55.68	317.3	26.44	26' 5 1/4"
8' 6"	102	8171.3	56.75	320.4	26.70	26' 8 3/8"
8' 7"	103	8332.3	57.82	323.6	26.95	26' 11 1/8"
8' 8"	104	8499.9	59.04	326.7	27.24	27' 2 7/8"
8' 9"	105	8659.0	60.13	329.9	27.49	27' 5 7/8"
8' 10"	106	8824.7	61.24	333.0	27.74	27' 8 7/8"
8' 11"	107	8992.0	62.49	336.2	28.00	28' 0"
9' 0"	108	9160.9	63.62	339.3	28.27	28' 3 1/4"
9' 1"	109	9331.3	64.75	342.4	28.53	28' 6 3/8"
9' 2"	110	9503.3	66.01	315.6	28.80	28' 9 5/8"
9' 3"	111	9676.9	67.20	348.7	29.06	29' 0 3/4"
9' 4"	112	9852.0	68.37	351.9	29.31	29' 3 3/4"
9' 5"	113	10028.8	69.69	355.0	29.59	29' 7 1/8"
9' 6"	114	10207.0	70.88	358.1	29.85	29' 10 1/4"
9' 7"	115	10386.9	72.08	361.3	30.10	30' 0 1/8"
9' 8"	116	10568.3	73.44	364.4	30.38	30' 4 1/2"
9' 9"	117	10751.3	74.66	367.6	30.63	30' 7 1/2"
9' 10"	118	10935.9	75.89	370.7	30.88	30' 10 1/2"
9' 11"	119	11122.0	77.29	373.9	31.16	31' 1 7/8"
10' 0"	120	11309.7	78.54	377.0	31.42	31' 5"
10' 1"	121	11499.0	80.12	280.1	31.70	31' 8 3/8"
10' 2"	122	11689.9	81.17	383.3	31.94	31' 11 1/4"
10' 3"	123	11882.3	82.51	386.4	32.20	32' 2 3/8"
10' 4"	124	12076.3	84.00	389.6	32.50	32' 6"
10' 5"	125	12271.9	85.22	392.7	32.56	32' 6 3/4"
10' 6"	126	12469.0	86.59	395.8	32.98	32' 11 3/4"
10' 7"	127	12667.7	87.96	399.0	33.25	33' 3"
10' 8"	128	12868.0	89.35	402.1	33.50	33' 6"
10' 9"	129	13069.8	90.75	405.3	33.77	33' 9 1/4"
10' 10"	130	13273.2	92.17	408.4	34.03	34' 3/8"
10' 11"	131	13478.2	93.6	411.6	34.30	34' 3 1/2"
11' 0"	132	13684.8	95.03	414.7	34.56	34' 6 3/4"
11' 1"	133	13892.9	96.48	417.8	34.81	34' 9 3/4"
11' 2"	134	14102.6	97.96	421.0	35.08	35' 1"
11' 3"	135	14313.9	98.70	424.1	35.34	35' 4 1/8"
11' 4"	136	14526.7	100.90	427.3	35.60	35' 7 1/4"
11' 5"	137	14741.1	102.36	430.4	35.87	35' 10 1/2"
11' 6"	138	14957.1	103.9	433.5	36.12	36' 1 1/2"
11' 7"	139	15174.7	105.3	436.7	36.40	36' 4 7/8"
11' 8"	140	15393.8	106.9	439.8	36.65	36' 7 7/8"
11' 9"	141	15614.5	108.4	443.0	36.90	37' 10 7/8"
11' 10"	142	15836.8	109.9	446.1	37.17	37' 2"
11' 11"	143	16060.6	111.5	449.3	37.44	37' 5 1/4"
12' 0"	144	16286.0	113.1	452.4	37.70	37' 8 1/2"
12' 1"	145	16513.0	114.6	455.5	37.96	37' 11 1/2"
12' 2"	146	16741.6	116.2	458.7	38.22	38' 2 5/8"
12' 3"	147	16971.7	117.8	461.8	38.49	38' 5 7/8"
12' 4"	148	17203.4	119.4	465.0	38.75	38' 9"
12' 5"	149	17436.6	121.0	468.1	39.01	39' 0 1/8"
12' 6"	150	17671.5	122.7	471.2	39.26	39' 3 1/8"

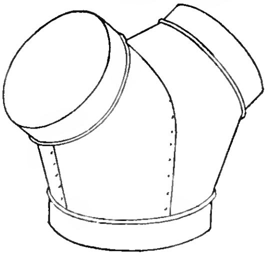
If the piping is made much lighter, particularly in the larger sizes, it will not keep its shape when laid horizontally, thereby decreasing the area and increasing the friction. Ordinarily rectangular pipes are made of the same gage metal as a round pipe of equivalent area.

Rectangular ducts for horizontal distribution require bracing, made of light angle iron, to



Branch and Reducer

FIG. 52.



"Y" Branch Connection

FIG. 53.

prevent sagging even with the heavier gages. In calculating the weight of piping 10 to 25 per cent should be added for laps and rivets.

MODERN PRACTICE IN SHOP AND FACTORY HEATING

The following is an extract from a paper on *Shop and Factory Heating* presented before the *Detroit Engineering Society, March, 1914,* by *F. R. Still:*

Those who recognize the advantages of a good heating plant frequently do not realize the necessity for designing it when the plans of the building are being made. The result is, that instead of arranging other things of much less importance to accommodate the heating plant, the latter has to be contorted, cut down, or squeezed into whatever space is available after everything else is taken care of. This makes the plant either expensive to install or to operate, and often it is so contorted or dwarfed in one way or another as to fail to give the satisfactory results naturally expected.

When it comes to laying out a heating plant, the character of the building and the purpose for which it is built must be carefully studied. What serves very nicely in a building of a certain type, in which a particular class of work is done, proves anything but satisfactory in a building of the same type in which a different kind of work is done.

Machine Shop and Paper Mill Compared. This can be made clear by comparing a machine shop with a paper mill. They are similar in size, shape and exposure. In the machine shop no steam or moisture is emitted to condense on the walls and roof; hence many of them are heated in a fairly satisfactory manner by direct radiation, consisting usually of steam coils strung around the outside walls. Such a system would never do in a paper mill, where every day several tons of water are thrown off into the atmosphere in the form of steam from the drying cylinders over which the paper passes. This steam must be taken up by a copious circulation of warm, dry air, or it will drip from the roof and run down the walls.

Again, in a shoe factory, shirt, glove, cap or other manufacturing plant where hundreds of employees are assembled in one room, the air very soon becomes unfit to breathe; it produces a dull, languid feeling, which saps all the vitality and ambition of the employees, makes them inert, forgetful, careless, subject to severe colds and other disabilities, all tending to make them irregular in their attendance to their duties. To overcome this the building must be well ventilated, and the ventilation of a building in the winter time must be considered in connection with the heating system.

Conditions in Foundries. Foundries are much like the paper mills just cited, as the steam rising from the moulds when pouring off, as well as the smoke and noxious gases, should be re-

moved. Roof ventilators, monitor lights or ventilating sashes are very poor makeshifts indeed. Perhaps they will let some of the smoke escape, and with it some of the gases and most of the heat; but most of the gases being heavier than the air lie in the lower strata near the floor and are not affected by the roof vents. Further, instead of removing the steam, they only accelerate the annoyance from it, as the warm air which escapes through the roof must be supplanted by outside air, and this, not being properly heated, chills the air within the building, thus reducing its capacity to carry moisture, thereby causing greater precipitation than there would be were no attempt made to ventilate the building.

Tendency of Modern Construction. Then, again, apart from the use to which the building is put, consideration must be given to its character.

The tendency nowadays is to put up lofty buildings. Such structures have steel frames, walls made almost entirely of glass, with only narrow pilasters between, and shallow panels of brick or concrete above and below the openings. The exposures are relatively enormous, requiring large amounts of heating surface, yet the spaces available for steam coils are limited to such an extent that they are often insufficient for the heating surface required.

Furthermore, when the space is sufficient, the banking of such great heating surfaces into a small space makes it uncomfortable for the employees who have to work near them.

It is these and many other features that have made the blower system popular, as it has proved efficacious in almost every building of whatever character, when it has not been misapplied.

Arrangement of Ducts in Industrial Plants. The prevailing practice in industrial plants is to make the ducts of galvanized sheet iron for all types of buildings, except where it is necessary to place them underground, in which case they are built of brick, vitrified tile, or reinforced concrete.

It is always advisable to run the ducts inside the building because of the heat wasted if they are run outside, be they above or below the ground. It is, furthermore, advisable to make the ducts of galvanized iron and suspend them in the open air overhead, if possible, as the loss of heat from underground ducts is enormous.

The average modern shop building has a trussed roof, the bottom chords of which are often 30 to 50 ft. above the floor. The ducts have to rest on these trusses, and it is impossible to drive air from them down to the floor in a way that will produce satisfactory results. This method has been tried often enough, but there still remain others who must be convinced of its impracticability by trying it themselves. The only way to obtain an even distribution of heat is to discharge heated air at such points as it is most needed and where the effect will be most appreciated.

To distribute the heat evenly necessitates running the ducts to all cold spots; it is needed the most in the lower strata near the floor, not up among the roof trusses; the greatest benefit is derived from the system by discharging the warm air close to the floor, keeping the lower strata in circulation and thereby warming it by mixing with the warm air discharged from the ducts.

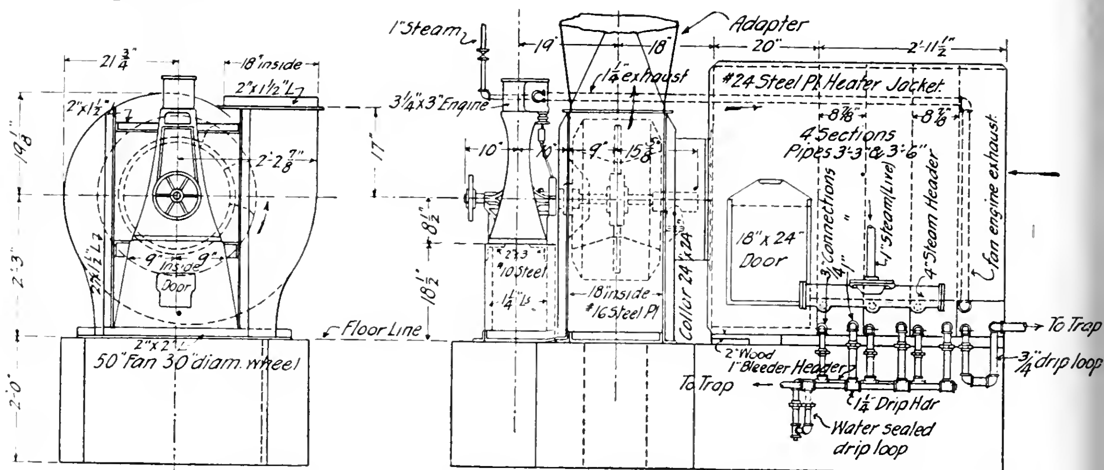
The best way to bring this about is to extend the branch ducts from the main trunk line over to the walls or to columns not more than 20 ft. away from the outside walls, then down toward the floor, ending 4 or 5 ft. from the floor. The air should discharge directly toward the floor or at only a slight angle from perpendicular. This method will be found most effective in machine shops, foundries and other lofty structures.

Different Arrangement Necessary Where Condensation is Likely to Collect on Under Side of Roof. In paper mills, rubber works, dye houses and other plants for which the building is of the same type as those just noted, it is necessary to blow some hot air out toward the roof as well as down toward the floor, in order to take care of the condensation which would otherwise collect on the under side of the roof. Even then, in very cold climates, it is sometimes necessary to put in a false ceiling to overcome this annoyance, unless hoods are provided to remove the vapor immediately, and particularly if the roof is built of a material which is a good conductor of heat.

For buildings which are several stories in height, each story being from 10 to 16 ft. high, the treatment should be different. With them it is possible, and sometimes advisable, to introduce the air near the ceiling, blowing downward at an angle of 30° to 45° from horizontal.

Frequently buildings of this character are quite effectively heated from one or two galvanized iron stand pipes run up through the middle of the building, with outlets into each story. This method is practical in buildings not over 60 ft. wide; if the building is not over 100 ft. long, one riser will be sufficient.

Warm Air Flues on Outside Face of Pilasters. For cotton, woolen, silk and other textile mills, it has become almost the universal practice to build vertical warm air flues on the outside face of the pilasters, on both sides of the building. These flues usually have a 2-in. air space built into the brickwork to insulate them. The air is admitted to each story about 8 ft. above the floor. Deflecting "Mill" dampers regulate the volume of air discharged through each opening. The various flues can be supplied at their base from a main duct built either of masonry or galvanized iron.



Draw Through Combination of S.P. Fan & Pipe Coil Heater.

FIG. 54.

For manufacturing plants, it is customary to make the trunk line ducts of such an area as will convey the required volume of air at a velocity varying from 1500 to 2400 ft. per minute. In high buildings used for heavy and coarse work, where most of the employees stand or move about considerably, the velocity can be much higher than in shops divided into several stories or those in which the work is more or less sedentary, like the manufacture of shirts, gloves, etc., where the employee sits all day, simply feeding the material into machines.

Air currents or drafts are of no material moment in the former shops, while in the latter they will produce great discomfort. Therefore, the latter class should have the main ducts of sufficient area to keep the velocity down to 1200 to 1500 feet per minute, and the branches should be proportioned to a velocity of 600 to 900 ft.

Another advantage the blower system possesses, infrequently brought to notice, is the cooling and comforting effect it has in oppresively warm weather in summer time. Simply running the fan will, of itself, greatly relieve the oppressiveness, and when cold water is circulated through the coils the difference is very noticeable.

Fan Driving

Fan Driving. The fan used in connection with hot blast apparatus is perhaps more often driven by a *steam engine* either *direct connected or belted*, the exhaust from the engine being turned

into one section of the heater, see Fig. 54. In this case the cost of operating the fan is practically nothing, as heat of the exhaust is fully utilized in the heater. (See " Exhaust Steam Heating.")

The size of the engine cylinder required is readily determined from the data given under " Steam Engines " when the horsepower, speed, and initial steam pressure are known.

Fans are driven by *motors* either *direct connected, or by means of belts or silent chain drives.* When direct connected a variable speed motor is advisable, but is somewhat more expensive. A belt or chain driven installation is a much cheaper combination owing to the fact that a high-speed motor may be used, which for the same output is smaller than a medium or low speed-motor.

When belts are used for the drives the pulley centers are placed ordinarily 5 to 6 feet on centers in order to obtain a sufficient contact on the driving pulley. The horsepower of belt drives will be found under the heading " Miscellaneous Tables." The horsepower capacity refers to an arc of contact of 180° on the driver, and if less, which is the usual case, the powers given must be reduced a proportionate amount. Silent chain drives are convenient for use in places where the space is limited, as the distance between the sprocket centers may be made comparatively small. See " Miscellaneous Tables " for chain drive data.

It is customary to figure either form of drive for an efficiency of 95 per cent. See chapter on " Electrical Equipment " for rating and dimensions of direct current motors.

APPLICATION OF HOT BLAST HEATING DATA

The application of the foregoing hot blast heating data to the design of a heating system for a modern shop building as shown by Fig. 55 follows.

The heat loss calculations given are based on maintaining an inside temperature of 60° F. when the outside temperature is 0°.

The infiltration loss is based on the estimated amount of air leakage around the perimeter of the steel ventilating sash in the side walls and the top hung continuous sash, and calculated by this method is approximately 70 per cent of the amount obtained by assuming the infiltration as one air change per hour in this particular case.

Heater Calculations. Assume the steam pressure maintained in the heater is 5 lb. gage and the velocity through the clear area of the heater to be 1200 ft. per min. measured at 70° F. Assume also a temperature of 145° F. for the air at duct outlets, and a temperature drop of 10° in the duct system. Temperature of air leaving heater to be 145 + 10 or 155°.

Refer to diagram Fig. 27 giving the temperature rise of air passed through pipe coil heaters at the velocity stated, we have:

27 rows 1″ pipe will raise the air from 0° to 154°
7 rows 1″ pipe will raise the air from 0° to 62°
20 rows 1″ pipe will raise the air from 62° to 154°

The *temperature of the air leaving* this 5 section heater will therefore be approximately 155° as assumed.

The weight of air to be circulated per hour is:

$$M = \frac{H}{0.24(t_d - t)} = \frac{1,423,920}{0.24(145 - 60)} = 70,000 \text{ lb. (nearly)}$$

The required *free area of the heater* is:

$$A = \frac{70,000}{60 \times 0.075 \times 1200} = 12.96 \text{ sq. ft.}$$

The nearest size section from Table 5 is 5′ – 0″ x 6′ – 6″. Total lineal ft. 1″ pipe for five sections is, 5 × 742 = 3710 ft., or 5 × 272 or 1360 sq. ft. of *heating surface.*

The *loss in pressure* through the heater, 1″ pipe on 2⅝″ centers, from Table 17, is 0.35″ water.

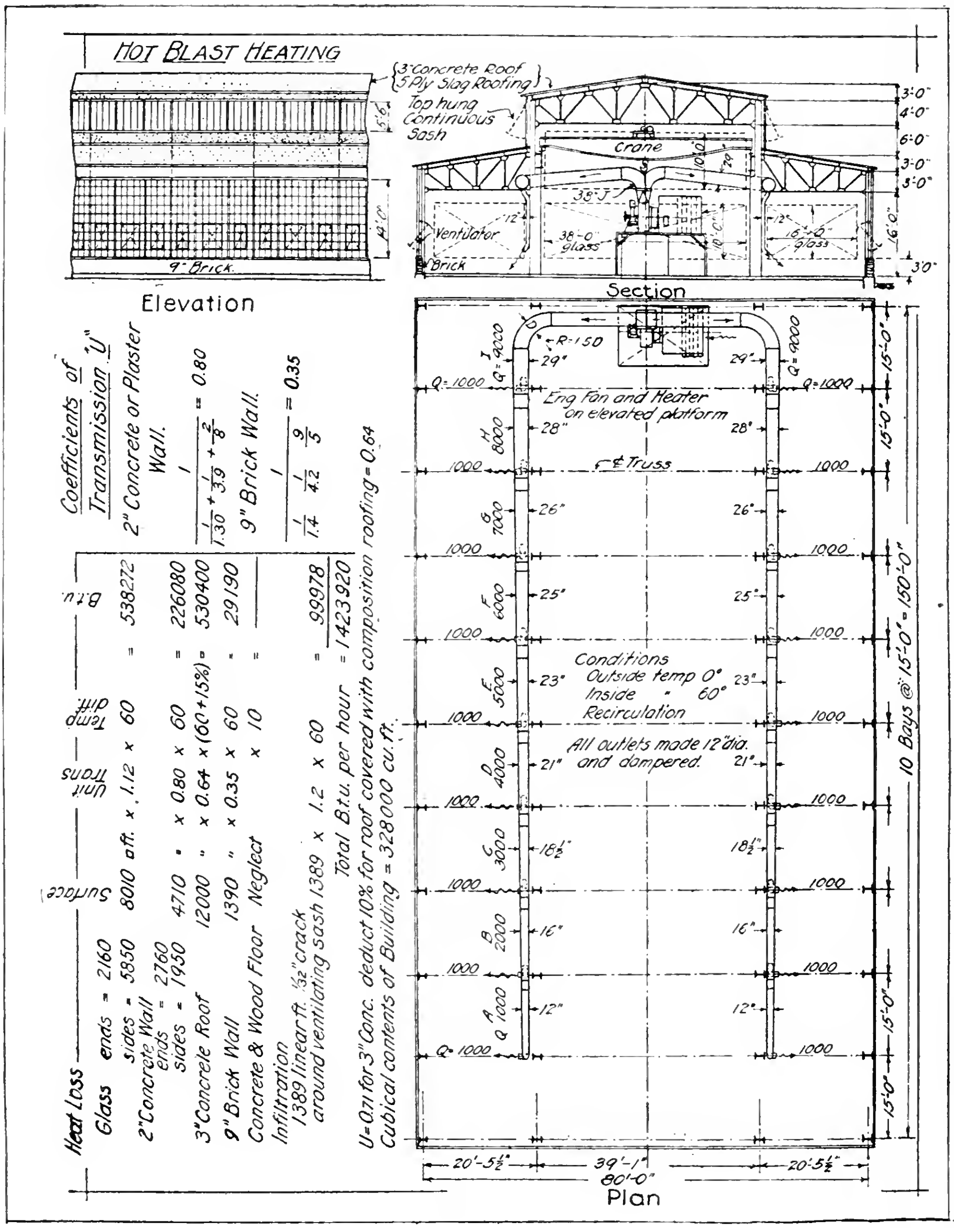

HOT BLAST HEATING
3" Concrete Roof
5 Ply Slag Roofing
Top hung Continuous Sash
Crane
Ventilator
9" Brick
Elevation
Section
Plan
Eng Fan and Heater on elevated platform
¢ Truss
Conditions
Outside temp 0°
Inside " 60°
Recirculation
All outlets made 12" dia. and dampered.
Heat Loss — Surface — Unit Trans — Temp diff. — B.t.u.
Glass ends = 2160 sides = 5850 8010 sq.ft. × 1.12 × 60 = 538272
2" Concrete Wall ends = 2760 sides = 1950 4710 " × 0.80 × 60 = 226080
3" Concrete Roof 12000 " × 0.64 × (60+15%) = 530400
9" Brick Wall 1390 " × 0.35 × 60 = 29190
Concrete & Wood Floor Neglect × 10 =
Infiltration 1389 linear ft. 1/32" crack around ventilating sash 1389 × 1.2 × 60 = 99978
Total B.t.u. per hour = 1423920
U = 0.71 for 3" Conc. deduct 10% for roof covered with composition roofing = 0.64
Cubical contents of Building = 328000 cu. ft.
Coefficients of Transmission 'U'
2" Concrete or Plaster Wall.
9" Brick Wall.
= 0.80
= 0.35
10 Bays @ 15'-0" = 150'-0"
20'-5½"
39'-1"
80'-0"

FIG. 55.

Weight of Steam Required per Hour. The weight of steam condensed per hour in the heater, 5 lb. gage pressure, will be:

$$W = \frac{0.24 \times 70{,}000 \times 156 - 60)}{962.2 \text{ (latent heat at 5 lb. gage)}} \text{ or } 1675 \text{ lb.}$$

Design of Duct. The round ducts will be designed for equal friction pressure loss per foot of length. The final velocity at the last or most remote outlet from fan will be taken as 1275 ft. per min. See Table 32. The friction pressure loss for this velocity, as read from chart Fig. 38, is 0.15″ water per 100 ft. of length for a value of $f = 0.0037$. There are to be 18 outlets. The total volume of air to be discharged measured at 145° F. is $70{,}000/(60 \times 0.065) = 18{,}000$ cu. ft. per min. or $18{,}000/18 = 1000$ cu. ft. per min. per outlet.

Area of outlet or last section is 1000/1275 sq. ft., corresponding to a diameter of 12″. The branch outlets may all be made the same size and provided with dampers to adjust or equalize the flow, or we may proportion the size of each branch from the total pressure available in the main at the point where the branch is taken off. The total pressure available in the main cannot be ascertained, however, until after the main has been designed, as previously shown.

TABLE 32

Section	Quantity of Air, Cubic Feet per Minute	Diameter, Inches	Velocity, Feet per Minute	Measured Length + Allowance for Ells
A	1000	12	1275	25′ + 1 × (6 + 3) = 34′
B	2000	16	....	15′
C	3000	18½	....	15′
D	4000	21	....	15′
E	5000	23	....	15′
F	6000	25	....	15′
G	7000	26	....	15′
H	8000	28	....	15′
I	9000	29	....	35′ + 2.4 (6 + 10) = 73′
J	18000	38	2285	....

Total length = 212′

The friction pressure loss in the duct system is therefore [212/100 × 0.15] + 50 per cent, or 0.48″ water.

Selection of Fan for "Draw Through" Arrangement. The total pressure required, referred to a temperature of 70°, is:

Pressure loss in heater (Data from Table 17, 1″ pipe 2⅝″ centers)	= 0.35″
Pressure loss in duct (Data from chart Fig. 38)	= 0.48″
Final velocity pressure (Data from Diagram Fig. 31)	= 0.11″
Total pressure	= 0.94″

The actual total pressure will be somewhat less, owing to the fact that the air temperature is higher (145° F.) and the density less than for air at 70° F. The actual estimated total pressure is therefore: $0.94 \times \frac{0.065 \text{ (dens. at } 145°)}{0.075 \text{ (dens. at } 70°)}$ or 0.81″. If a Sirocco fan is to be used the required static pressure rating is 0.81/1.15 or 0.704″ (say ¾″).

The volume of air the fan must handle in this example is 18,000 cu. ft. per min., measured at 145° F. As stated under "Effect of Temperature on Fan Performance," to maintain a constant pressure the tabulated speed, volume, and pressure must be multiplied by the square root

of the ratio of densities or $\sqrt{\frac{0.075}{0.066}} = 1.07$ (nearly), Table 27.

We therefore locate in fan Table 24 a size of fan having a capacity, measured at 70° F., equal to $\frac{18,000}{1.07}$ or 16,822 cu. ft. per min. (say 17,000) when operating with a static pressure of ¾″.

A No. 8 Sirocco fulfills this requirement. The tabulated speed and horsepower when multiplied by the factor 1.07 gives: $196 \times 1.07 = 210$ r. p. m. and $3.76 \times 1.07 = 4.02$ brake horsepower.

Selection of Fan for "Blow Through" Arrangement. In this case the fan may be called upon to handle air at a temperature of 0° F. or lower. Assuming the same weight of air, or 70,000 lb. per hour, to be handled by the fan at a static pressure of ¾″ the volume at 0° is $\frac{70,000}{0.086 \times 60} =$ 13,566 cu. ft. per min. Referring to Table 27 the ratio between the speed, volume and power necessary to produce the same pressure for air at 0° and air at 70° is found to be 0.932. We therefore choose a fan having a capacity of 13,566 0.932 or 14,557 cu. ft. at 70° and ¾″ static pressure.

Referring to the performance curves, Fig. 48, we find that a No. 7 fan at 240 r. p. m. or a No. 9 fan at 195 r. p. m. fulfills the requirement.

Selection of Motor for Fan Driving. It is considered good practice to add 15 to 25 per cent to the brake horsepower, as determined from the fan tables, for the rating of the motor to allow for a possible overload due to the fact that the fan may not be operated under exactly the same conditions as to pressure and speed for which it was originally rated.

For the preceding example (blow through arrangement) a 5 horsepower motor would be selected.

Additional Heating Requirement. It is frequently desirable to proportion the heating apparatus large enough so that the fan may be shut down at night and started up about two hours before the shop or factory is opened in the morning.

In this event we may safely assume that the temperature of the air in the building will not be below 30° F. when the fan is started and that the air is all recirculated.

The fan and heater must be of sufficient capacity to take care of the heat loss from the building, including the infiltration, and in addition warm up the contained air from 30° to 60°, in two hours. Assuming the same data as given in the preceding example, the additional heat required will be: $\frac{328,000 \times 0.08 \times 0.24 \times 30}{2} = 94,464$ B.t.u. per hour.

This amounts to an increase of approximately 7 per cent in the heating requirements as previously calculated, and is readily provided for by increasing the steam pressure carried in the heater to approximately 10 lb. gage.

TYPICAL HOT BLAST INSTALLATIONS

The use of indirect radiators and fans for supplying air both for heating and ventilating has been discussed in detail in the preceding pages and a typical open factory layout (Fig. 55) has been given, including the calculations for same. The following plans show the most common methods of applying these systems to buildings of several floors where the ventilation and heating requirements for the various rooms may be quite different:

Single Duct System for School Using Plenum Chamber. The principal features of the heating and ventilating system of the *John Greusel School, Detroit, Mich.*, are shown in Fig. 56. This building is 78′ x 233′ over all and covers an area of about 14,000 sq. ft. It is 2 stories high above the basement, which is 3′ – 6″ below grade and 9′ – 0″ high in the clear except

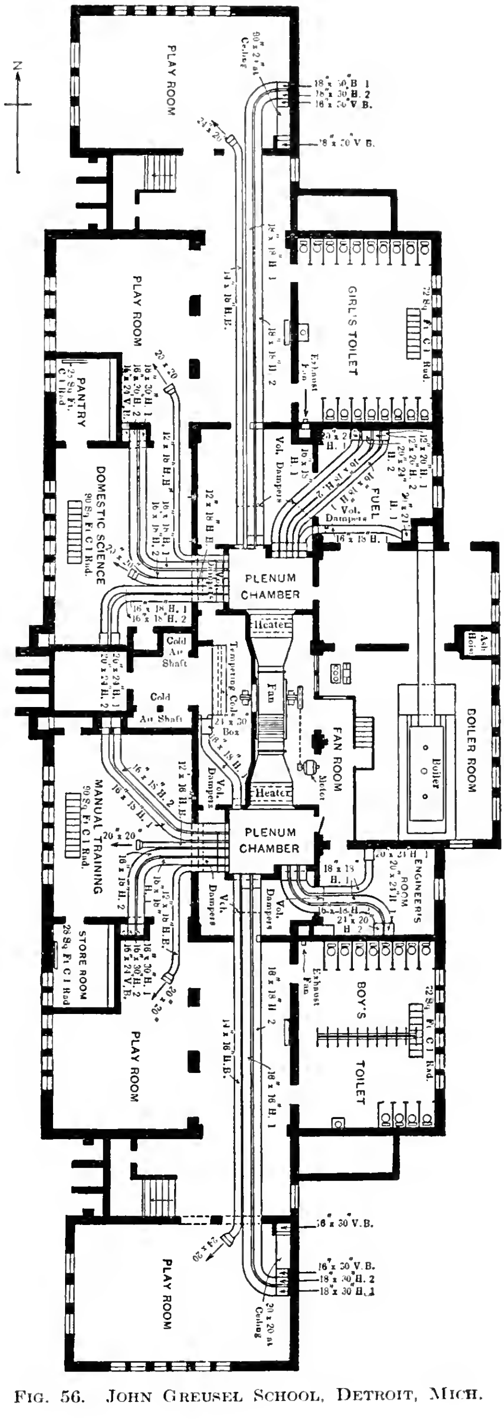
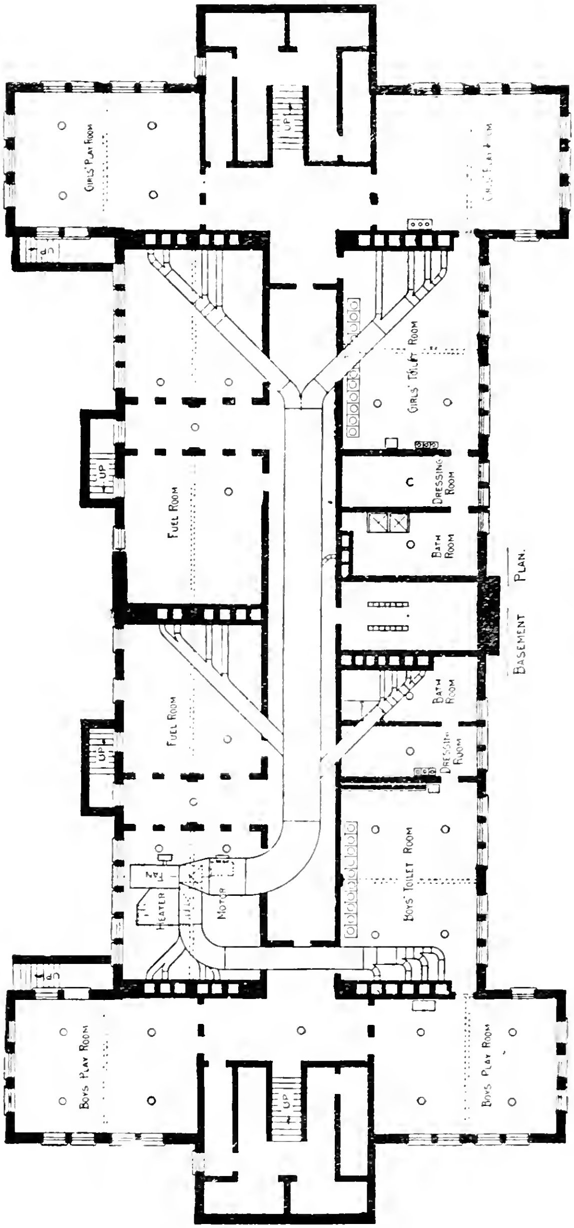


FIG. 56. JOHN GREUSEL SCHOOL, DETROIT, MICH.



Basement of Large School Fitted with Green Heating and Ventilating Apparatus

IG. 57. TRUNK DUCT SYSTEM FOR SCHOOLS

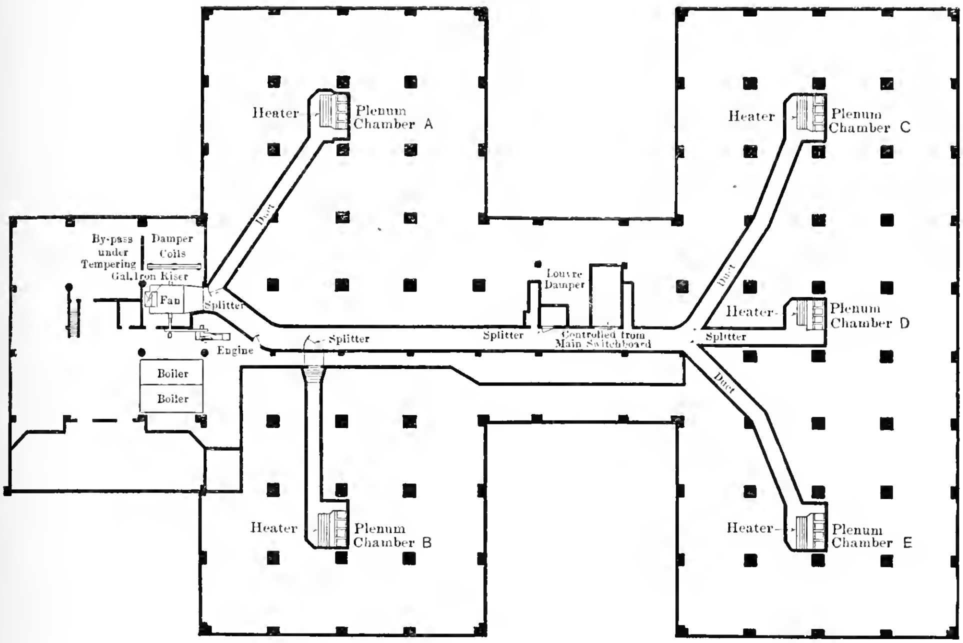


FIG. 58. BASEMENT PLAN.

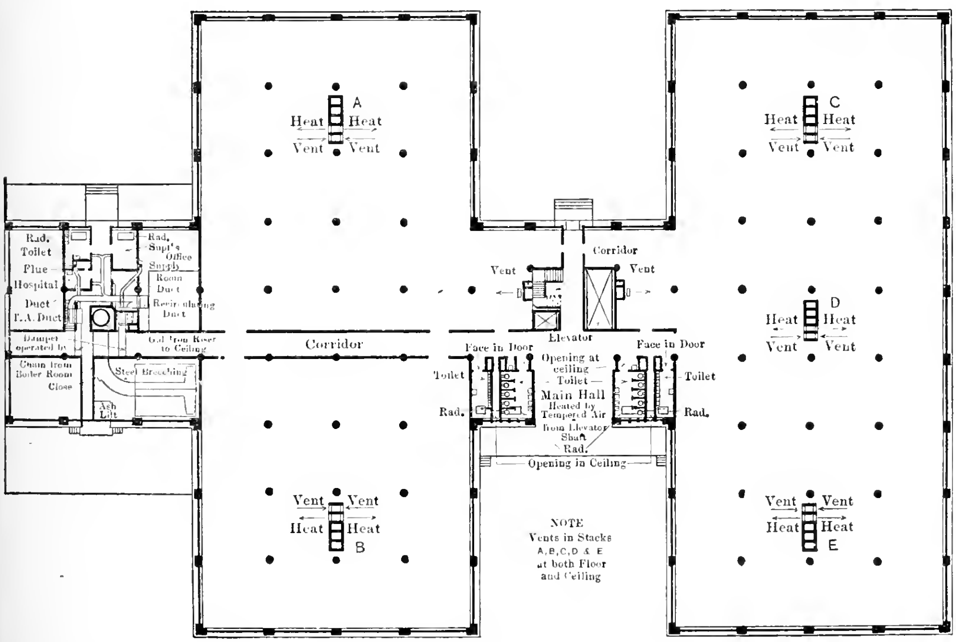


FIG. 59. TYPICAL FLOOR PLAN.

for fan and boiler rooms, which are 10′ – 0″ and 14′ – 0″ respectively. The way in which the basement space is utilized is shown on plan.

The first floor is 13′ – 0″ in the clear and contains eight school rooms, kindergarten room

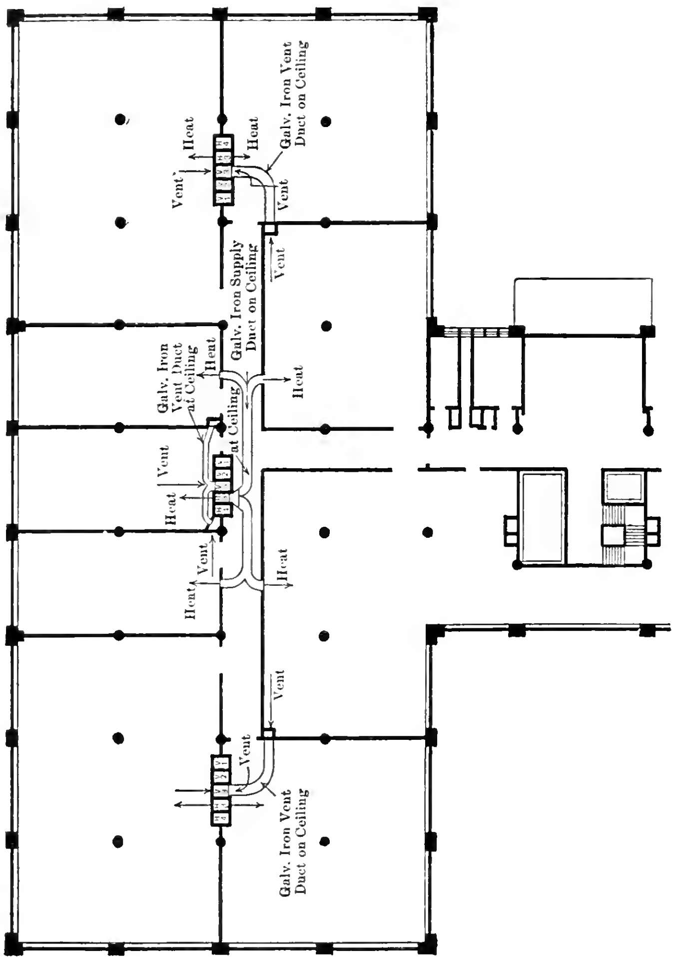


FIG. 60. THIRD FLOOR PLAN.

and office. The second floor is 13′ – 3″ in the clear and contains eight school rooms, recitation room, library and teachers' rest room.

The air supplied by the fan in this building is used for both heating and ventilating, so that it must enter above room temperature, but at the same time *some direct radiation* is located in the more exposed rooms and used only in very severe weather. Direct radiation only is used in the corridors and storerooms, and air only is supplied play-rooms.

The *steel plate fan* (*American Blower Co.* No. 160) is 8 ft. in diameter and 38″ wide at periphery with double top horizontal discharge, and is belted to a 20 horsepower Wagner single-phase motor. The fan runs at 180 r. p. m. and requires 15 horsepower when delivering 41,000

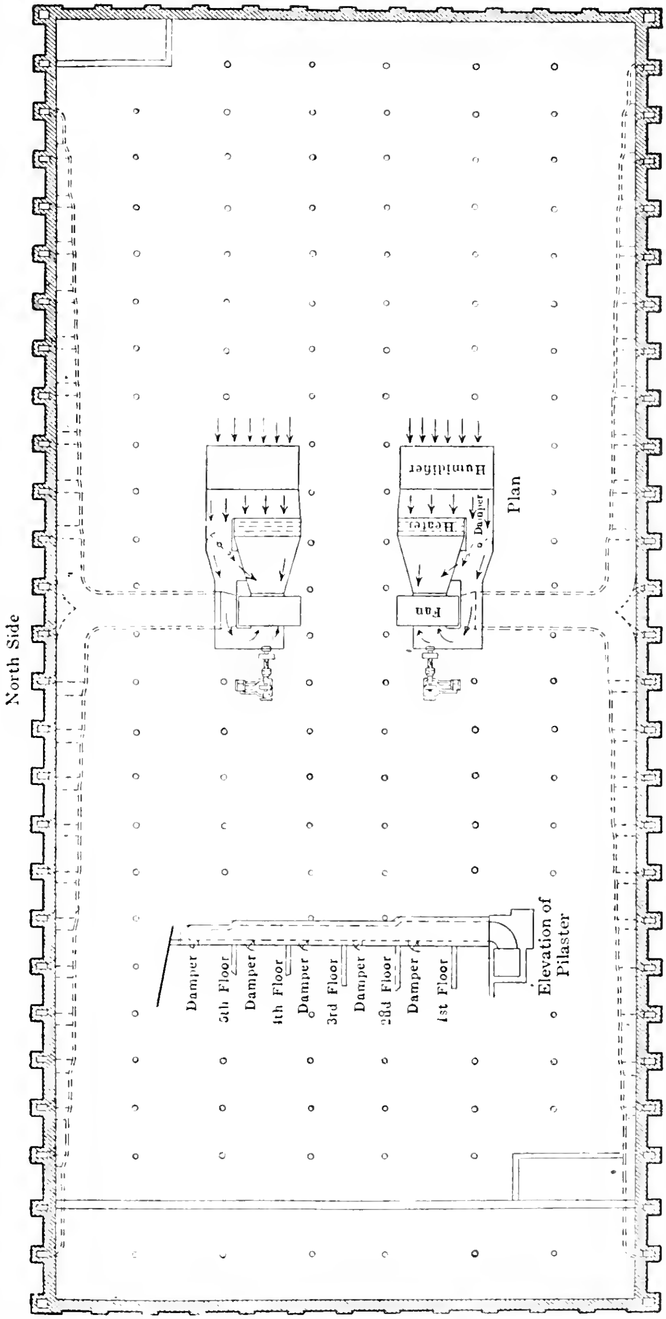


FIG. 61. FACTORY SYSTEM WITH FAN AND HEATER IN BASEMENT. DUCTS IN OUTSIDE PILASTERS.

cu. ft. of air at 90° F. per minute to the building. This fan must always run at rated speed, using alternating current. The air supply is taken at the attic floor line through lowered openings, but by the use of suitable dampers in the attic it is possible to recirculate the air in the building when it is not occupied.

The *heating and tempering coils* are 10 and 8 rows deep respectively. The former are in

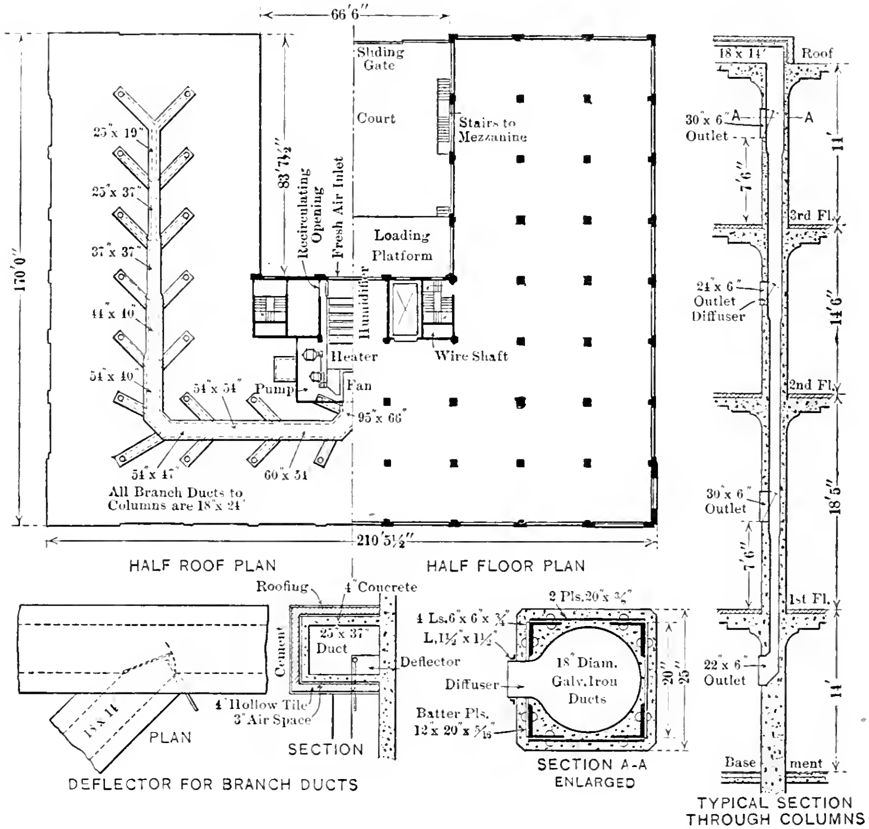


FIG. 62. FACTORY SYSTEM WITH FAN AND HEATER ON ROOF. DUCTS IN HOLLOW COLUMNS.*

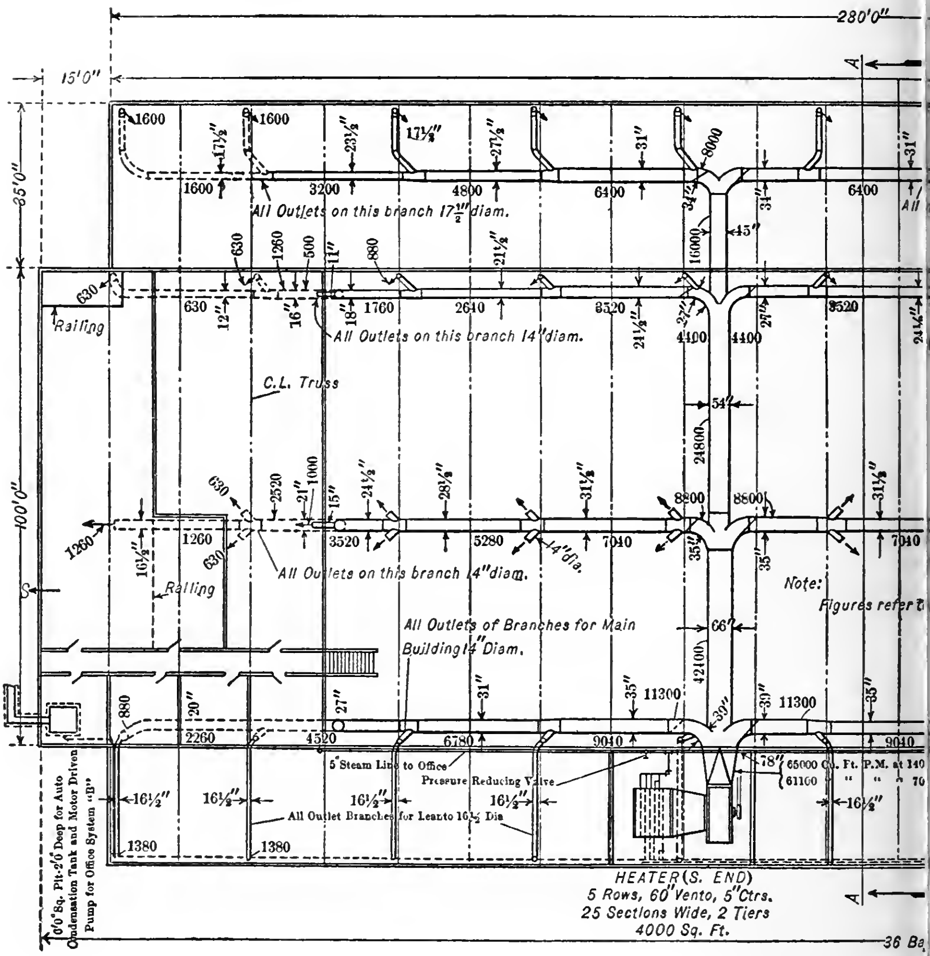
two banks and contain 2790 lineal ft. of 1" pipe each while the latter contains 1900 lineal ft., a grand total of 7480 lineal ft. for the entire system.

The *horizontal return tubular boiler* is 72" in diameter, 16'–0" long and contains 86 3½" tubes. A *Murphy* automatic stoker set in a *Dutch* oven is used, and the horizontal grate area is 36 sq. ft., more than required except in starting up with building cold. The steam pressure is maintained at from 10 to 20 lb. gage, both stoker engine and feed pump operating on from 7 to 10 lb. gage steam pressure.

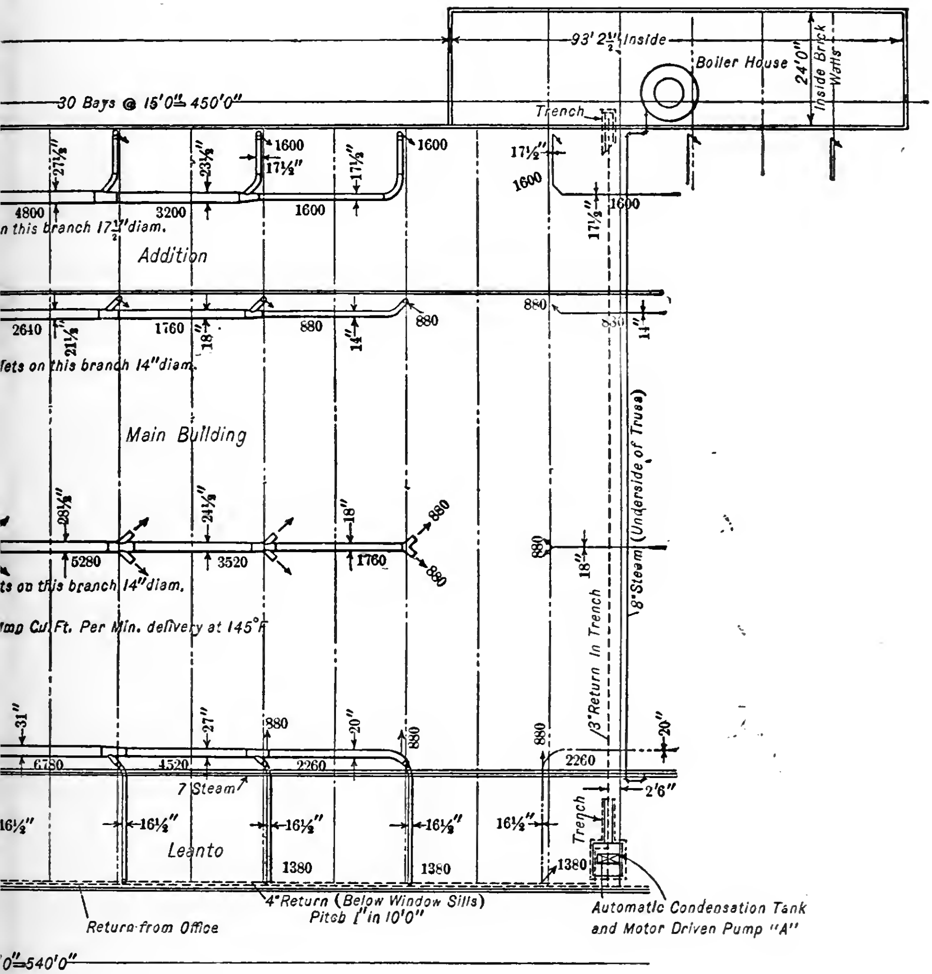
The *condensation pump and its receiver* handle only radiation returns, as its exhaust steam is wasted to the atmosphere to avoid trouble from oily exhaust. The stoker engine exhausts into stoker grates. The pump is duplex, 7½" steam cylinder, 3" water cylinder and 6" stroke. The boiler is also equipped with a low pressure injector and city water connection.

The *steam piping system* is valved in such a way as to permit of operating either the tem-

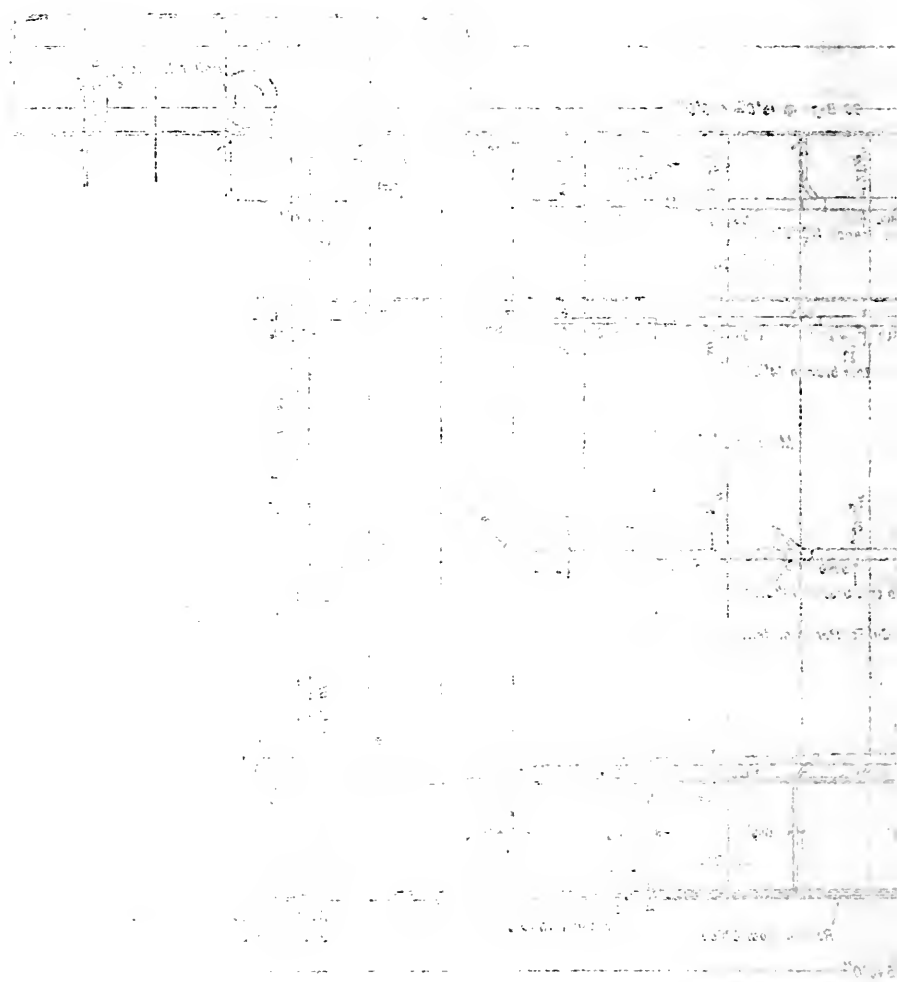
*From *Eng. News*.



280'0"
15'0"
35'0"
100'0"
All Outlets on this branch 17½" diam.
All Outlets on this branch 14" diam.
C.L. Truss
All Outlets on this branch 14" diam.
Railing
Railing
All Outlets of Branches for Main Building 14" Diam.
Note:
Figures refer
5" Steam Line to Office
Pressure Reducing Valve
All Outlet Branches for Leanto 16½ Dia
65000 Cu. Ft. P.M. at 140
61100
6'0" Sq. Pit-2'6" Deep for Auto Condensation Tank and Motor Driven Pump for Office System "B"
HEATER (S. END)
5 Rows, 60" Vento, 5" Ctrs.
25 Sections Wide, 2 Tiers
4000 Sq. Ft.
36 Ba



93' 2½" Inside
Boiler House
24'0" Inside Brick Walls
30 Bays @ 15'0" = 450'0"
Trench
n this branch 17½" diam.
Addition
fets on this branch 14" diam.
Main Building
ts on this branch 14" diam.
mp Cu. Ft. Per Min. delivery at 145°F
8" Steam (Underside of Truss)
8" Return In Trench
7 Steam
Leanto
4" Return (Below Window Sills) Pitch 1" in 10'0"
Return from Office
Automatic Condensation Tank and Motor Driven Pump "A"
0" = 540'0"



pering coils, heating coils, direct radiation or auxiliary apparatus independently and without filling the entire system with steam. All piping is covered.

There are 39 *direct radiators* containing 2400 sq. ft. of surface, and are thermostatically controlled by the *Johnson* system except in corridors and store room.

The *temperature control system* is also applied to the mixing dampers and tempering coils, and pneumatically controlled recirculating and fresh air dampers are installed in the attic.

Individual *air supply ducts* are run from the plenum chambers to each room, and each duct has a volume damper near the plenum chamber. These *horizontal ducts* are usually 16″ x 18″ but for the long rooms ducts 18″ x 18″ are used.

The *vertical flues* are in general 20″ x 24″ for each room. Diffusing vanes are used at the stack heads, and the registers for these flues are 24″ x 24″.

The *actual air delivery* was found to be 1750 cu. ft. per min. at the register or 35 cu. ft. per min. per pupil. The following velocities result: (1) air outlet = 437′ per min.; (2) vertical flues = 530′ per min.; (3) horizontal ducts = 875′ per min. It will be seen that velocities at room outlets are one-half the velocity in basement ducts.

The *vent flues* are connected into the top of the wardrobes built into the side walls of each school room. These wardrobes are 7 ft. long and the sliding doors stop about 7″ above the floor. These flues are connected into four main groups, each of which is connected to a 54″ roof ventilator.

No air is supplied to the toilets, and each room has an 18″ exhaust fan discharging into a separate system of vent flues connected to independent roof ventilators.

The *total cost of the heating and ventilating plant*, including stoker, temperature regulation, and electric wiring (electric light wiring included) was $14,591. Since the electric light wiring cost approximately $2200, the heating and ventilating plant proper cost $12,391.

Trunk Duct System for Schools. Where air is to be supplied for ventilation only a trunk duct system (Fig. 57) may be used, as it is possible to control the temperature of the entire air supply at one point and discharge all the air at the same temperature for each room. The sizes for the system shown are not given as the description of the single duct plenum chamber system will serve to cover the general field of school house heating and ventilating.

Factory Heating and Ventilating System with Reheaters. A recent and somewhat special type of factory heating system is shown in Figs. 58, 59 and 60 as applied to the new *Toledo Factories Company* building. In this system underground basement ducts are used to carry tempered air to centrally located reheater chambers directly at the base of the vertical heating and ventilating flues. In this way the heat loss from the underground main ducts is practically eliminated, and regulation of the air temperature for each floor or room easily maintained.

The arrangement of the main heat and vent flues is such that either a whole floor (Fig. 59) or a subdivided floor (Fig. 60) may be served with a minimum of supply and vent ducts.

Factory Heating Systems without Reheaters. One of the earliest successful systems for heating mills is shown in Fig. 61. In this case the fan and heater of the "draw through" type are placed in the basement, and the vertical risers from the main duct are carried up the outside walls of the building in hollow pilasters.

A much more compact system of factory heating is made possible by placing the fan and heater on the roof and running the drop risers down to the various floors through the hollow interior columns as shown in Fig. 62. This latter system is decidedly more efficient than the former, as the heat loss from the distributing duct system, usually at high temperature, is materially reduced in this case, and the actual length of air travel is very much shorter.

Factory Heating System Using Cast-Iron Down-Draft Boilers. An unusual factory heating system of large proportions, in which steam is supplied by cast-iron boilers of the smokeless or down-draft type, is shown in Figs. 63 to 68.

This system was designed by one of the authors in accordance with the data given in the text, the duct system being designed for equal friction pressure loss per foot of length. All sizes are given on the plans, which also show the important details of the system.

The boiler plant is remarkable in that nine cast-iron down-draft *Ideal* boilers of the largest type are installed, making this the largest single installation of this equipment in the country.

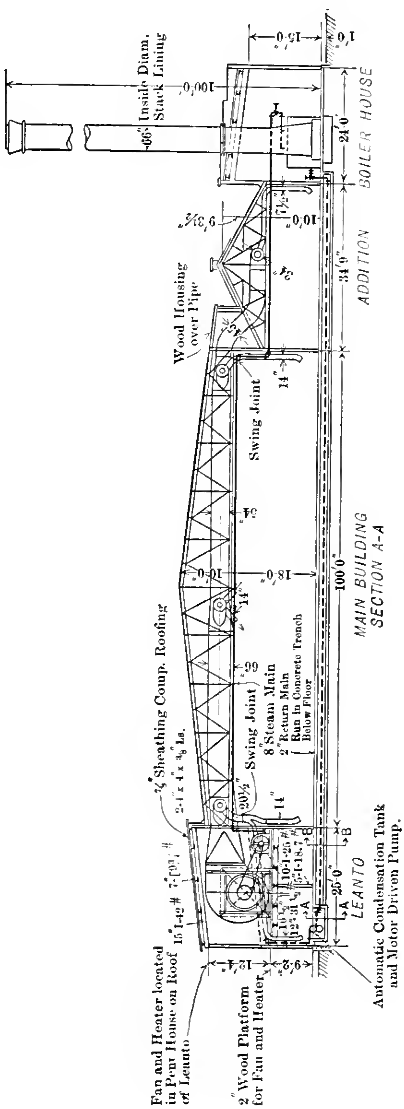
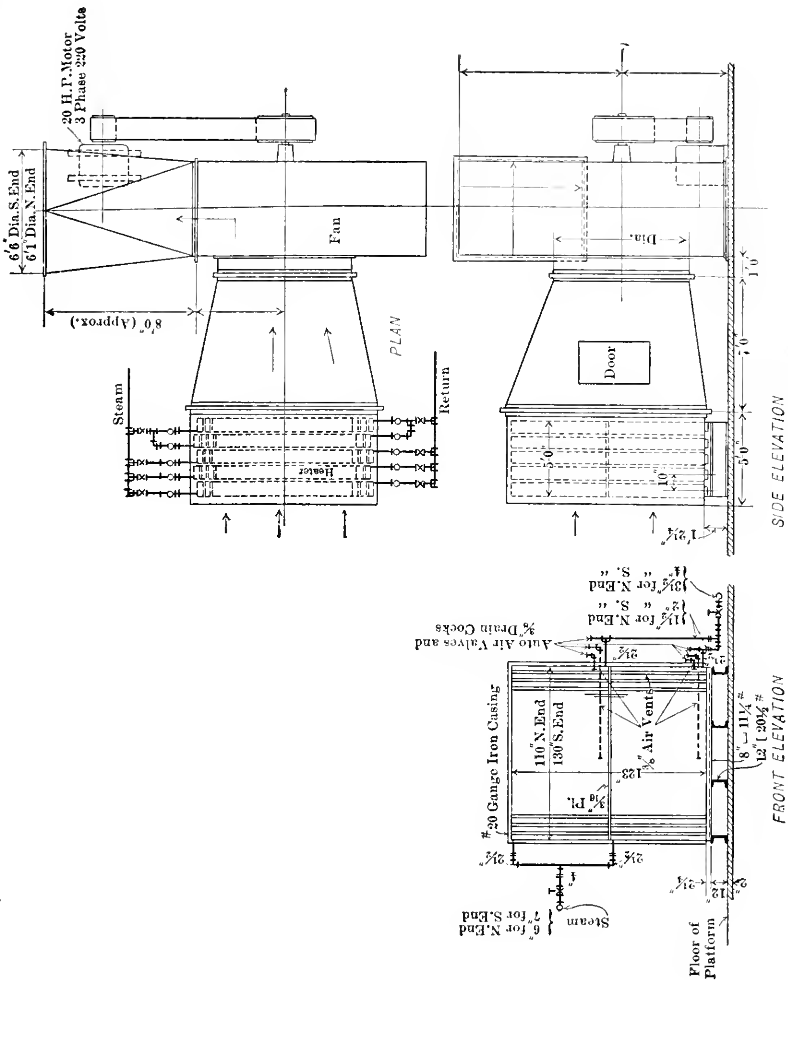


FIG. 64.



FAN AND HEATER ARRANGEMENT

FIG. 65

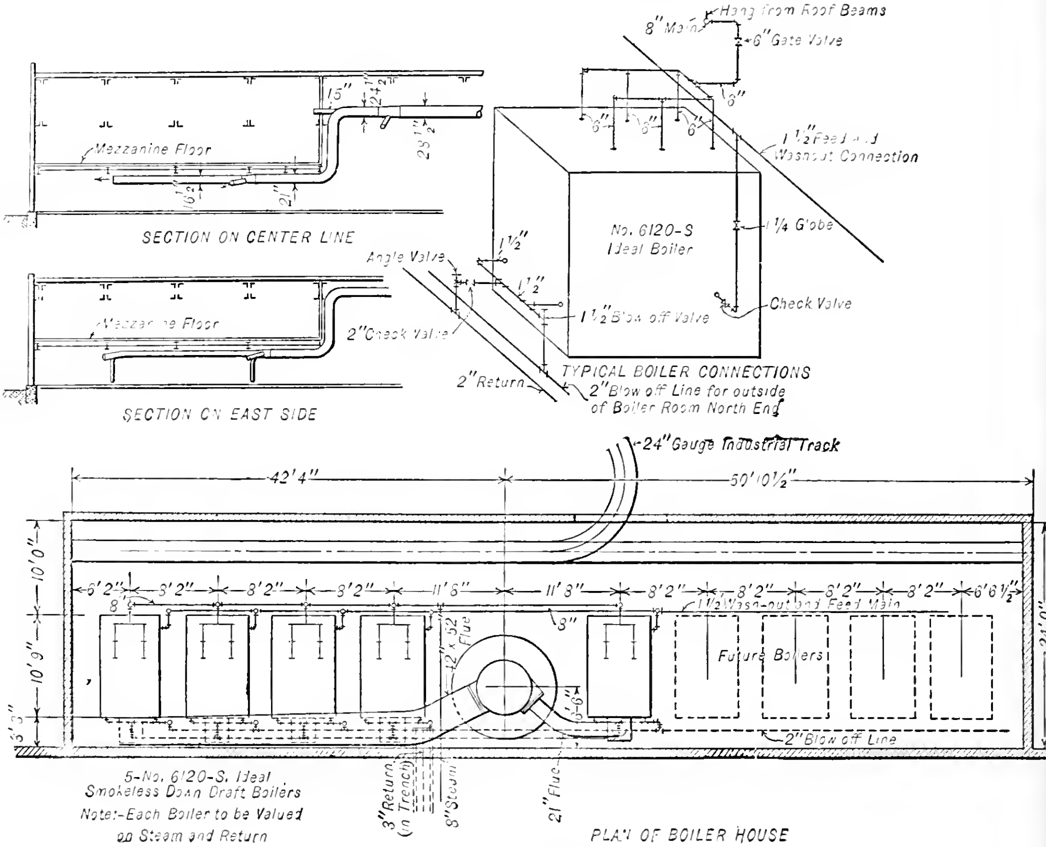


FIG. 66.

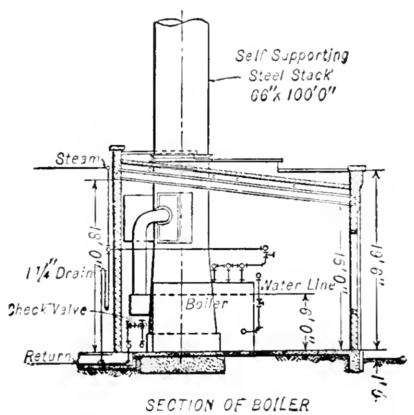


FIG. 67.

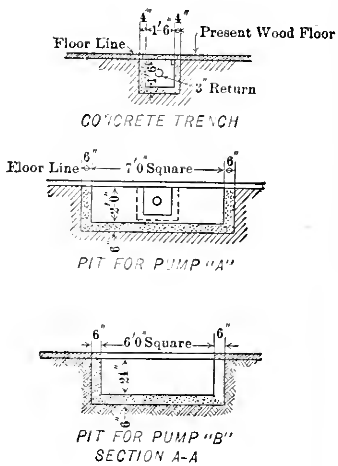


FIG. 68.

CHAPTER XVI

AIR CONDITIONING, AIR WASHING, HUMIDIFYING, COOLING, AND DRYING

HUMIDITY

Definition. Humidity is the water vapor (steam or moisture) mixed with the air.

The maximum weight of vapor which a given enclosure will contain is dependent only upon the temperature (see Steam Tables) regardless of the presence or absence of any other vapor **or** gas. That is, the weight of vapor is exactly the same whether the air is present or not.

Dalton's Law. Each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature, had no other gas or vapor been present. If p = the observed pressure of the mixture and p_1 p_2, p_3, etc., = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc.}$$

Saturated Air. Air is said to be *saturated* when it has mixed with it the maximum possible amount of vapor, the amount of which varies with the temperature. The vapor itself under this condition is also saturated (quality $x = 1$). If the air is not in a saturated condition, then the contained vapor is in a superheated state.

The *actual humidity* of the air, in meteorological work, is the number of grains (1 lb. = 7000 grains) or pounds of water vapor contained by one cu. ft. of a mixture of air and vapor at the observed temperature.

The *relative humidity* or *degree of humidity* is the percentage of ratio of the actual amount of moisture (grains or lb.) contained by one cu. ft. of the mixture to the amount which one cu. ft. of the mixture would hold at the same temperature, if saturated. The condition is stated as so many per cent relative humidity.

It simplifies calculations somewhat, if the actual humidity is considered as the *number of pounds of saturated vapor mixed with one pound of dry air, when saturated at a given temperature and pressure*, and the relative humidity, as the *weight of vapor actually mixed with one pound of air divided by the amount of saturated vapor mixed with one pound of air when saturated at the same temperature and pressure*, and expressed as a percentage.

Example. Let it be required to find the weight of vapor carried by one pound of air in a saturated mixture of air and vapor at a temperature of 60° F. and atmospheric pressure (14.7 lb. per sq. in. absolute at sea level).

If p_1 = Absolute partial pressure of the vapor lb. per sq. in. corresponding to the temperature. (See Steam Tables.)

p_2 = Absolute partial pressure of the air lb. per sq. in.

p = Total or barometric pressure = (14.7 lb. per sq. in. absolute at sea level, or 29.92 in. of mercury).

$p = p_1 + p_2 = 14.7$ at sea level.

From the steam tables (saturated water vapor) for a temperature of 60° F.,

$p_1 = .26$ and density $= 0.00082$ lb. per cu. ft.

$p_2 = 14.70 - 0.26 = 14.44$ partial air pressure.

From the relation $PV = MRT$, (Law for perfect gases).

Where R for air $= 53.35$, $T = 459.6 + 60$, $P = 144 \times 14.44$, $M = 1$ lb.,

$$V = \frac{53.35 \times 519.6}{144 \times 14.44} = 13.33 \text{ cu. ft. volume of the air.}$$

This also is the volume of the saturated vapor, as the air and vapor occupy the same amount of space. The weight of the saturated vapor is therefore:

13.33×0.00082 or 0.01093 lb. per lb. of the air in the mixture.

The weight of vapor per cu. ft. of the mixture is $0.01093/13.33 = 0.00082$ lb. or $0.00082 \times 7000 = 5.74$ grains. The density of the mixture (1 lb. of air and its vapor) is $1.0109/13.33$ or 0.0758 lb. and its specific volume is $1/0.0758$ or 13.18 cu. ft.

Formula for Saturated Air. (100 per cent Relative Humidity.) The operation in the previous problem may be expressed by a formula as follows:

t = Temperature of the mixture degrees F.
T = Absolute temperature = $(t + 459.6)$.
P = Barometric pressure lb. per sq. ft.
P_s = Absolute vapor pressure lb. per sq. ft. corresponding to temperature t.
P_a = Absolute air pressure lb. per sq. ft.
$P = P_s + P_a$ and $P_a = P - P_s$.
V = Specific volume of air (cu. ft. per lb.) at temperature t.
V_s = Specific volume of saturated vapor at temperature t.
D_s = Density of saturated vapor at temperature t.
W = Weight of saturated vapor per pound of dry air in the mixture = $V D_s$.

Then $P_a V = RT = 53.35\ (t + 459.6)$.

$$V = \frac{53.35\ (t + 459.6)}{P_a} = \frac{53.35 + (t + 459.6)}{P - P_s}$$

$$\text{and } W = \frac{53.35\ (t + 459.6)\ D_s}{P - P_s}$$

$$\text{or } W = \frac{0.37\ (t + 459.6)\ D_s}{P - P_s}$$ when the pressures are stated in pounds per sq. inch.

If the weight is stated in grains and the pressures in inches of mercury,
1 lb. = 7000 grains, 1 in. mercury = 70.721 lb. per sq. ft.

$$\text{Then } G = \frac{5284\ (t + 459.6)\ D_s}{P_m - P_n}$$

in which G = grains moisture per lb. of dry air, P_m = barometric pressure inches of mercury and P_n = absolute pressure of saturated water vapor corresponding to the temperature, in inches of mercury. See Table 1 for "Properties of Saturated Air," also "Heat Exchange Diagram" (Fig. 2).

Dew Point Temperature. The temperature corresponding to saturation (100 per cent relative humidity) for a given weight of vapor is known as the *dew point*.

Any lowering of the temperature produces a contraction of volume and a partial condensation, the amount of vapor condensed being the difference between the original amount and the amount carried at saturation for the new or lower temperature. Air with any amount of vapor has a "dew point," as the temperature can always be lowered so that condensation must take place.

The maximum amount of saturated vapor which may be mixed with air in forming a *saturated mixture* may be calculated by making use of *Dalton's* law of partial vapor pressures.

The total pressure (barometric pressure) of a mixture of air and vapor is made up of the sum of the partial vapor pressure (vapor tension) and the partial air pressure.

Adiabatic Saturation of Dry Air. If absolutely dry air is passed through an insulated chamber containing a sponge, saturated with water, Fig. 0, it is observed that the temperature of the water will be lowered until a stationary temperature t' is reached, which is lower than the temperature t of the incoming air. Furthermore, the temperature of the leaving air will be the same as the temperature of the water.

It is evident that an exchange of heat must take place between the air and water, as heat is neither supplied by or extracted from an external source. A heat transfer of this sort is said to be *adiabatic.*

The evaporation of the water takes place at the recorded temperature of the liquid t'.

Let W = weight of water evaporated per lb. of dry air passed through the apparatus, determined by actual measurement.

r' = latent heat of saturated vapor corresponding to temperature of liquid or t'.

Then $0.2411\ (t - t')$ = B.t.u. given up by one pound of air,

$r'W$ = heat required to evaporate the weight of moisture added to the air.

$r'W = 0.2411\ (t - t')$ which is the equation for the adiabatic saturation of dry air.

If the experiment were performed with dry air having an initial temperature $t = 75°$ the observed temperature of the water would be $t' = 46°$ and the weight of water evaporated per lb. of dry air, by measurement, $W = 0.00656$ lb. The latent heat for 46° is $r' = 1065.6$; then

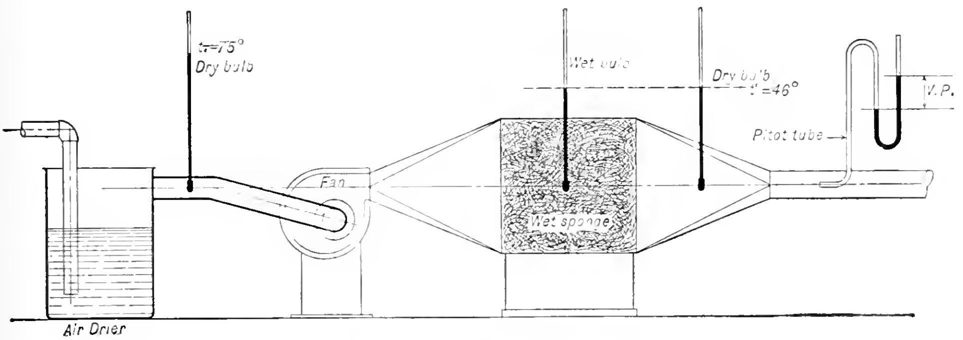


FIG. 0. ADIABATIC SATURATION OF DRY AIR.

the heat required for evaporation is 1065.6×0.00656 or 6.99 B.t.u. which is seen to be exactly the same as the heat given up by the pound of dry air or $0.2411 \times (75 - 46)$ or 6.99 B.t.u.

Adiabatic Saturation of a Mixture of Air and Vapor. Assume (Fig. 3) a saturated mixture of 1 lb. of dry air plus W_1 lb. of vapor corresponding to temperature t_1 as obtained above corresponding to condition (2). If the temperature of this mixture is now raised to t corresponding to condition (3) the vapor is superheated.

The mixture will become adiabatically saturated at temperature t' corresponding to condition (4).

The heat given up by the superheated mixture of 1 lb. of air plus W_1 lb. of vapor in having its temperature lowered from t to t' is

$$C_{pa}\ (t - t') + C_{ps}\ W_1\ (t - t')\ \text{B.t.u.}$$

$$C_{ps} = \text{Sp. heat of vapor at constant pressure.}$$

If W' is the weight of vapor in a saturated mixture at temperature t' then the weight of vapor added to saturate the mixture adiabatically is $(W' - W_1)$. And the heat required for evaporation is $r'\ (W' - W_1)$. As this is an adiabatic change, no heat supplied from an external source, the following equality exists:

$$r'\ (W' - W_1) = C_{pa}\ (t - t') + C_{ps}\ W_1\ (t - t').$$

The constant weight of vapor lines are plotted by adding the heat required to raise the temperature of the mixture from saturation t_1 to the required temperature t. Thus for condition (3) add $C_{pa}\ (t - t_1) + C_{ps}\ W_1\ (t - t_1)$ to the B.t.u. in 1 lb. of saturated air above 0° at temperature t_1, condition (2).

TABLE 1
MIXTURES OF AIR AND SATURATED WATER VAPOR
G. A. GOODENOUGH

Temp., °F.	* Pressure of saturated vapor		Weight of saturated vapor				Volume in cu. ft.		Heat content in B.t.u. of 1 lb of dry air above 0° F.	Latent heat of vapor, B.t.u.	† Heat content in B.t.u. of 1 lb. of dry air with vapor to saturate it
	In. of Hg.	Lb. per sq. in.	per cu. ft. Pounds	per cu. ft. Grains	per lb. of dry air Pounds	per lb. of dry air Grains	of 1 lb. of dry air	of 1 lb. of dry air + vapor to saturate it			
0	0.0375	0.0184	0.0000674	0.472	0.000781	5.47	11.58	11.59	0.0	0.964	0.964
2	.0417	.0204	.0000746	.522	.000869	6.08	11.63	11.65	0.482	1.071	1.553
4	.0462	.0227	.0000823	.576	.000963	6.74	11.68	11.70	0.964	1.186	2.150
6	.0512	.0252	.0000909	.636	.001067	7.47	11.73	11.75	1.446	1.313	2.759
8	.0567	.0279	.0001001	.701	.001183	8.28	11.78	11.80	1.928	1.455	3.383
10	0.0628	0.0308	0.0001103	0.772	0.001309	9.16	11.83	11.86	2.411	1.608	4.019
12	.0694	.0341	.000121	.850	.001447	10.13	11.88	11.91	2.893	1.776	4.669
14	.0766	.0376	.000134	.935	.001599	11.19	11.94	11.97	3.375	1.961	5.336
16	.0846	.0415	.000147	1.028	.001764	12.35	11.99	12.02	3.858	2.162	6.020
18	.0932	.0458	.000161	1.128	.001946	13.62	12.04	12.08	4.340	2.383	6.723
20	0.1027	0.0504	0.000177	1.237	0.002144	15.01	12.09	12.13	4.823	2.623	7.446
22	.1130	.0555	.000194	1.356	.002360	16.52	12.14	12.19	5.305	2.885	8.190
24	.1242	.0610	.000212	1.485	.002596	18.17	12.19	12.24	5.787	3.170	8.957
26	.1365	.0670	.000232	1.625	.002854	19.98	12.24	12.30	6.270	3.482	9.752
28	.1499	.0736	.000254	1.776	.003134	21.94	12.29	12.35	6.752	3.821	10.573
30	0.1646	0.0809	0.000278	1.943	0.003444	24.11	12.34	12.41	7.234	4.195	11.429
32	.1806	.0887	.000303	2.124	.003782	26.47	12.39	12.47	7.716	4.058	11.783
33	.1880	.0923	.000315	2.206	.003938	27.57	12.41	12.49	7.96	4.22	12.18
34	.1957	.0961	.000327	2.292	.004100	28.70	12.44	12.52	8.20	4.40	12.60
35	0.2036	0.1000	0.000340	2.380	0.004268	29.88	12.47	12.55	8.44	4.57	13.02
36	.2119	.1041	.000353	2.471	.004442	31.09	12.49	12.58	8.68	4.76	13.44
37	.2204	.1083	.000367	2.566	.004622	32.35	12.52	12.61	8.93	4.95	13.87
38	.2292	.1126	.000381	2.663	.004809	33.66	12.54	12.64	9.17	5.14	14.31
39	.2384	.1171	.000395	2.764	.005002	35.01	12.57	12.67	9.41	5.35	14.76
40	0.2478	0.1217	0.000410	2.868	0.005202	36.41	12.59	12.70	9.65	5.56	15.21
41	.2576	.1266	.000425	2.976	.005410	37.87	12.62	12.73	9.89	5.78	15.67
42	.2678	.1315	.000441	3.087	.005625	39.38	12.64	12.76	10.14	6.01	16.14
43	.2783	.1367	.000457	3.201	.005848	40.93	12.67	12.79	10.38	6.24	16.62
44	.2891	.1420	.000474	3.319	.006078	42.55	12.69	12.82	10.62	6.48	17.10

* Below 32° F. the pressure of saturated vapor in contact with ice is given. † Values in this column do not include the heat of the liquid. Below 32° F. the heat of sublimation of ice is included.
Table based on 29.92" barometric pressure.

TABLE 1—(*Continued*)

MIXTURES OF AIR AND SATURATED WATER VAPOR

G. A. GOODENOUGH

Temp., °F.	Pressure of saturated vapor		Weight of saturated vapor				Volume in cu. ft.		Heat content in B.t.u. of 1 lb of dry air above 0° F.	Latent heat of vapor, B.t.u.	* Heat content in B.t.u. of 1 lb. of dry air with vapor to saturate it
	In. of Hg.	Lb. per sq. in.	per cu. ft.		per lb. of dry air		of 1 lb. of dry air	of 1 lb. of dry air + vapor to saturate it			
			Pounds	Grains	Pounds	Grains					
45	0.3003	0.1475	0.000492	3.442	0.00632	44.21	12.72	12.85	10.86	6.73	17.59
46	.3120	.1532	.000510	3.568	.00656	45.94	12.74	12.88	11.10	6.99	18.09
47	.3240	.1591	.000528	3.698	.00682	47.73	12.77	12.91	11.34	7.26	18.60
48	.3364	.1652	.000547	3.832	.00708	49.58	12.79	12.94	11.58	7.54	19.12
49	.3492	.1715	.000567	3.970	.00736	51.49	12.82	12.97	11.83	7.83	19.65
50	0.3624	0.1780	0.000588	4.113	0.00764	53.47	12.84	13.00	12.07	8.12	20.19
51	.3761	.1848	.000609	4.260	.00793	55.52	12.87	13.03	12.31	8.43	20.74
52	.3903	.1917	.000630	4.411	.00823	57.64	12.89	13.07	12.55	8.75	21.30
53	.4049	.1989	.000653	4.568	.00855	59.83	12.92	13.10	12.79	9.08	21.87
54	.4200	.2063	.000676	4.729	.00887	62.09	12.95	13.13	13.03	9.41	22.45
55	0.4356	0.2140	0.000699	4.895	0.00920	64.43	12.97	13.16	13.28	9.76	23.04
56	.4517	.2219	.000724	5.066	.00955	66.85	13.00	13.20	13.52	10.13	23.64
57	.4684	.2300	.000749	5.242	.00991	69.35	13.02	13.23	13.76	10.50	24.25
58	.4855	.2384	.000775	5.424	.01028	71.93	13.05	13.26	14.00	10.89	24.88
59	.5032	.2471	.000802	5.611	.01066	74.60	13.07	13.30	14.24	11.28	25.52
60	0.5214	0.2561	0.000829	5.804	0.01105	77.3	13.10	13.33	14.48	11.69	26.18
61	.5403	.2654	.000858	6.003	.01146	80.2	13.12	13.36	14.72	12.12	26.84
62	.5597	.2749	.000887	6.208	.01188	83.2	13.15	13.40	14.97	12.56	27.52
63	.5798	.2848	.000917	6.418	.01231	86.2	13.17	13.43	15.21	13.01	28.22
64	.6005	.2949	.000948	6.633	.01276	89.3	13.20	13.47	15.45	13.48	28.93
65	0.6218	0.3054	0.000979	6.855	0.01323	92.6	13.22	13.50	15.69	13.96	29.65
66	.6438	.3162	.001012	7.084	.01370	95.9	13.25	13.54	15.93	14.46	30.39
67	.6664	.3273	.001046	7.320	.01420	99.4	13.27	13.58	16.18	14.97	31.15
68	.6898	.3388	.001080	7.563	.01471	103.0	13.30	13.61	16.42	15.50	31.92
69	.7139	.3506	.001116	7.813	.01524	106.6	13.32	13.65	16.66	16.05	32.71
70	0.7386	0.3628	0.001153	8.069	0.01578	110.5	13.35	13.69	16.90	16.61	33.51
71	.7642	.3754	.001190	8.332	.01634	114.4	13.38	13.73	17.14	17.19	34.33
72	.7906	.3883	.001229	8.603	.01692	118.4	13.40	13.76	17.38	17.79	35.17
73	.8177	.4016	.001269	8.882	.01751	122.6	13.43	13.80	17.63	18.41	36.03
74	.8456	.4153	.001310	9.168	.01813	126.9	13.45	13.84	17.87	19.05	36.91

* Values in this column do not include the heat of the liquid.

TABLE 1—(*Continued*)

MIXTURES OF AIR AND SATURATED WATER VAPOR

G. A. GOODENOUGH

Temp., °F.	Pressure of saturated vapor		Weight of saturated vapor				Volume in cu. ft.		Heat content in B.t.u. of 1 lb of dry air above 0° F.	Latent heat of vapor, B.t.u.	* Heat content in B.t.u. of 1 lb. of dry air with vapor to saturate it
	In. of Hg.	Lb. per sq. in.	per cu. ft.		per lb. of dry air		of 1 lb. of dry air.	of 1 lb. of dry air + vapor to saturate it			
			Pounds	Grains	Pounds	Grains					
75	0.8744	0.4295	0.001352	9.46	0.01877	131.4	13.48	13.88	18.11	19.71	37.81
76	.9040	.4440	.001395	9.76	.01942	135.9	13.50	13.92	18.35	20.38	38.73
77	.9345	.4590	.001439	10.07	.02010	140.7	13.53	13.96	18.59	21.08	39.67
78	.9658	.4744	.001485	10.39	.02080	145.6	13.55	14.00	18.84	21.80	40.64
79	.9981	.4903	.001532	10.72	.02152	150.6	13.58	14.05	19.08	22.55	41.63
80	1.0314	0.5066	0.001580	11.06	0.02226	155.8	13.60	14.09	19.32	23.31	42.64
81	1.0656	.5234	.001629	11.40	.02303	161.2	13.63	14.13	19.56	24.11	43.67
82	1.1008	.5406	.001680	11.76	.02381	166.7	13.65	14.17	19.80	24.92	44.72
83	1.1370	.5584	.001732	12.12	.02463	172.4	13.68	14.22	20.04	25.76	45.80
84	1.174	.5767	.001786	12.50	.02547	178.3	13.70	14.26	20.29	26.62	46.91
85	1.212	0.5955	0.001841	12.89	0.02634	184.4	13.73	14.31	20.53	27.51	48.04
86	1.251	.6148	.001897	13.28	.02723	190.6	13.75	14.35	20.77	28.43	49.20
87	1.292	.6347	.001955	13.68	.02815	197.0	13.78	14.40	21.01	29.38	50.39
88	1.334	.6551	.002014	14.10	.02910	203.7	13.80	14.45	21.25	30.35	51.61
89	1.377	.6761	.002075	14.53	.03008	210.6	13.83	14.50	21.50	31.36	52.86
90	1.421	0.6977	0.002137	14.96	0.03109	217.6	13.86	14.55	21.74	32.39	54.13
91	1.466	.7200	.002201	15.41	.03213	224.9	13.88	14.60	21.98	33.46	55.44
92	1.512	.7427	.002267	15.87	.03320	232.4	13.91	14.65	22.22	34.59	56.78
93	1.560	.7660	.002334	16.34	.03430	240.1	13.93	14.70	22.46	35.69	58.15
94	1.609	.7901	.002403	16.82	.03544	247.1	13.96	14.75	22.71	36.86	59.56
95	1.659	0.8148	0.002474	17.32	0.03662	256.3	13.98	14.80	22.95	38.06	61.01
96	1.710	.8401	.002546	17.82	.03783	264.8	14.01	14.86	23.19	39.30	62.48
97	1.763	.8662	.002621	18.35	.03908	273.6	14.03	14.91	23.43	40.57	64.00
98	1.818	.8929	.002697	18.88	.04036	282.5	14.06	14.97	23.67	41.88	65.55
99	1.874	.9204	.002775	19.42	.04169	291.8	14.08	15.02	23.91	43.24	67.15
100	1.931	0.9486	0.002855	19.98	0.04305	301.3	14.11	15.08	24.16	44.63	68.79
101	1.990	0.9775	.002937	20.56	.04446	311.2	14.14	15.14	24.40	46.07	70.47
102	2.051	1.0072	.003021	21.15	.04591	321.4	14.16	15.20	24.64	47.54	72.18
103	2.113	1.0376	.003107	21.75	.04741	331.9	14.19	15.26	24.88	49.07	73.95
104	2.176	1.0689	.003195	22.36	.04895	342.7	14.21	15.33	25.13	50.64	75.77

* Values in this column do not include the heat of the liquid.

TABLE 1—(*Continued*)

MIXTURES OF AIR AND SATURATED WATER VAPOR

G. A. GOODENOUGH

Temp., ° F.	Pressure of saturated vapor		Weight of saturated vapor				Volume in cu. ft.		Heat content in B.t.u. of 1 lb of dry air above 0° F.	Latent heat of vapor, B.t.u.	* Heat content in B.t.u. of 1 lb. of dry air with vapor to saturate it
	In. of Hg.	Lb. per sq. in.	per cu. ft.		per lb. of dry air		of 1 lb. of dry air	of 1 lb. of dry air + vapor to saturate it			
			Pounds	Grains	Pounds	Grains					
105	2.241	1.1010	0.003285	22.99	0.0505	354	14.24	15.39	25.37	52.26	77.63
106	2.308	1.134	.003377	23.64	.0522	365	14.26	15.46	25.61	53.92	79.53
107	2.377	1.168	.003472	24.30	.0539	377	14.29	15.52	25.85	55.64	81.49
108	2.448	1.202	.003568	24.98	.0556	389	14.31	15.59	26.09	57.41	83.50
109	2.520	1.238	.003667	25.67	.0574	402	14.34	15.66	26.33	59.23	85.57
110	2.594	1.274	0.003769	26.38	0.0593	415	14.36	15.73	26.58	61.11	87.69
111	2.670	1.311	.003873	27.11	.0612	428	14.39	15.80	26.82	63.04	89.86
112	2.748	1.350	.003979	27.85	.0631	442	14.41	15.87	27.06	65.04	92.10
113	2.827	1.389	.004087	28.61	.0652	456	14.44	15.95	27.30	67.10	94.40
114	2.909	1.429	.004198	29.39	.0673	471	14.46	16.02	27.55	69.22	96.77
115	2.993	1.470	0.004312	30.18	0.0694	486	14.49	16.10	27.79	71.40	99.10
116	3.079	1.512	.004428	31.00	.0717	502	14.52	16.18	28.03	73.65	101.68
117	3.167	1.555	.004547	31.83	.0739	518	14.54	16.26	28.27	75.97	104.24
118	3.257	1.600	.004669	32.68	.0763	534	14.57	16.35	28.51	78.36	106.87
119	3.349	1.645	.004793	33.55	.0788	551	14.59	16.43	28.76	80.80	109.56
120	3.444	1.692	0.004920	34.44	0.0813	569	14.62	16.52	29.00	83.37	112.37
125	3.952	1.941	.005599	39.19	.0953	667	14.75	16.99	30.21	97.33	127.54
130	4.523	2.221	.006356	44.49	.1114	780	14.88	17.53	31.42	113.64	145.06
135	5.163	2.536	.007197	50.38	.1305	913	15.00	18.13	32.63	132.71	165.34
140	5.878	2.887	.008130	56.91	.1532	1072	15.13	18.84	33.85	155.37	189.22
145	6.677	3.280	0.00916	64.1	0.1800	1260	15.26	19.64	35.06	182.05	217.1
150	7.566	3.716	.01030	72.1	.2122	1485	15.39	20.60	36.27	214.03	250.3
155	8.554	4.201	.01156	80.9	.2511	1758	15.52	21.73	37.48	252.61	290.1
160	9.649	4.739	.01294	90.6	.2987	2091	15.64	23.09	38.69	299.55	338.2
165	10.860	5.334	.01445	101.1	.3577	2504	15.77	24.75	39.91	357.75	397.7
170	12.20	5.990	0.01611	112.8	0.4324	....	15.90	26.84	41.12	431.2	472.3
175	13.67	6.71	.01793	125.5	.5290	....	16.03	29.51	42.33	526.0	568.3
180	15.29	7.51	.01991	139.4	.6577	....	16.16	33.04	43.55	651.9	695.5
185	17.07	8.38	.02206	154.4	.8359	....	16.28	37.89	44.76	826.1	870.9
190	19.01	9.34	.02441	170.9	1.0985	....	16.41	45.00	45.97	1082.3	1128.3
200	23.46	11.53	0.02972	208.0	2.2953	....	16.67	77.24	48.40	2247.5	2296

* Values in this column do not include the heat of the liquid.

The Wet and Dry Bulb Psychrometer Principles Involved. The actual amount of moisture mixed with the air under various conditions of temperature and degrees of saturation is most conveniently ascertained by observing the *temperature at which evaporation takes place*, and the actual temperature of the air.

The temperature at which evaporation takes place is recorded by a thermometer, around the bulb of which is placed a moist cloth. This thermometer is termed the *wet bulb thermometer.*

If the spray water, through which air not initially saturated is passed, as in a humidifier, be simply recirculated and not supplied with heat from an external source in order to maintain its temperature constant, and having an initial temperature higher than that of the entering air, the temperature of the water will soon be lowered to that of the entering air. The water will then not be able to heat the air further, but will have its temperature lowered by any evaporation that may take place. The temperature of the water being lowered by evaporation, the cooled water will lower the temperature of the air, which, in turn, will give up some heat to the water by the reduction of its temperature. This heat exchange, between the air and water, will continue until a stationary water temperature (t') is reached, at which point the heat given up by the air to the water will just balance the heat required for evaporation. As no heat is supplied from an external source, it will be observed that this is an adiabatic change.

The air leaving is then in an adiabatically saturated condition, the temperature of which is that as recorded by the wet bulb thermometer, as the action described is similar to that which takes place when air is passed over the wet cloth of the wet bulb thermometer.

This furnishes a means for ascertaining the actual amount of moisture mixed with the air as given by the following method, devised by *W. H. Carrier:*

Psychrometric Method for the Determination of the Actual Weight of Moisture per Pound of Dry Air.

t = temperature of the air degrees F. (dry bulb).
t' = temperature of the air wet bulb. (This is the temperature at which the air becomes adiabatically saturated, *and not the dew-point temp.*)
$t - t'$ = wet bulb depression.
W = weight of moisture actually mixed with one lb. of dry air at temperature t.
$W' - W$ = weight of moisture per lb. dry air added in order to saturate the air.
r' = latent heat of vaporization at temperature t'.
$(W' - W)\,r'$ = heat necessary (B.t.u.) to evaporate $(W' - W)$ lb. water at temperature t'.
C_{ps} = Sp. heat of vapor at constant pressure (average value 0.44).
C_{pa} = Sp. heat of air at constant pressure (average value 0.24).

As this is an adiabatic change (no heat abstracted or added from an external source), the heat required for evaporation being supplied by the air and its contained vapor in lowering the temperature from t to t', then

$$(W' - W)\,r' = C_{ps}\,W\,(t - t') + C_{pa}\,(t - t')$$

$$W = \frac{r'\,W' - 0.24\,(t - t')}{r' + 0.44\,(t - t')}$$

The relative humidity is the ratio of W/W_x, W_x being the weight of moisture per lb. of air when saturated with vapor at temperature t. The *dew point* temperature is the temperature corresponding to saturated air containing W lb. of vapor per lb. of air in the mixture and should not be confounded with the *wet bulb temperature.*

The determination of the actual weight of vapor in one pound of dry air is most conveniently made by the use of *the wet and dry bulb sling psychrometer.*

This instrument (Fig. 1) consists of a wet bulb thermometer mounted adjacent to a dry bulb thermometer and so arranged that the entire mounting, which is about 15 inches in length, may be swung about a handle. In order to secure accurate and consistent results the instrument should be revolved from 150 to 225 times per min. For very accurate work *Carrier* states

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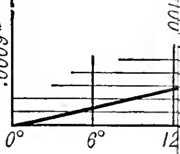
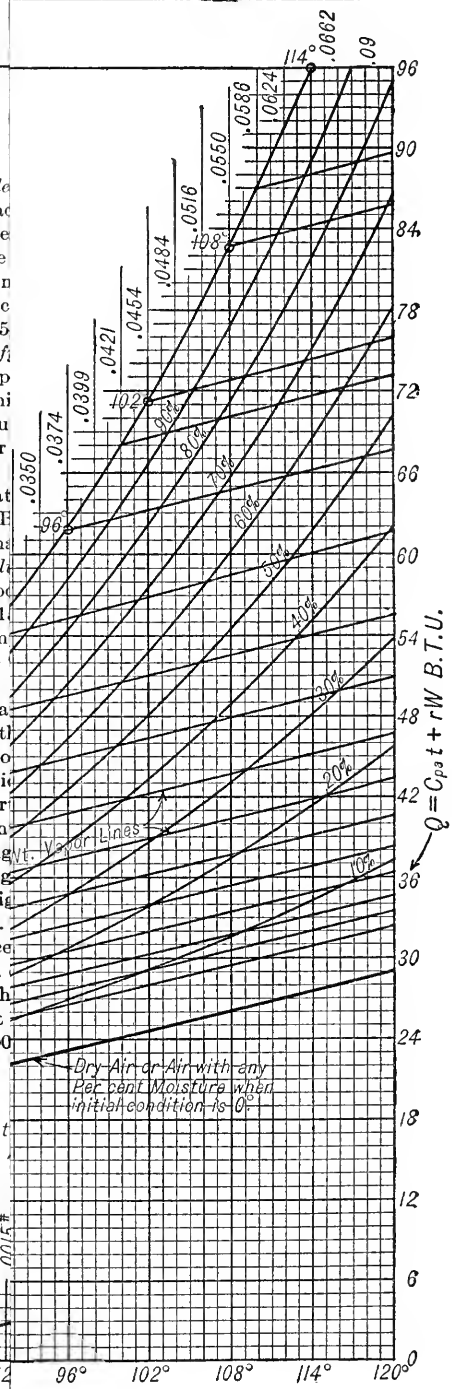




PSYCHROMETRIC CHART AND HEAT EXCHANGE DIAGRAM

L. A. Harding

Examples in the Use of Chart and Diagram. Required the relative humidity for a dry-bulb reading of 84° F. and a wet-bulb reading of 66° F.

The intersection of a horizontal line through 66° F. on the *saturation curve*, and the vertical through 84° F., dry bulb on the base line gives, approximately, 37 per cent for the relative humidity. The *dew point* temperature is found by tracing the corresponding *constant weight vapor line* to its intersection with the *saturation curve*, giving 55° F. for the above condition. The actual weight of vapor per pound of dry air may be read direct from the saturation curve for the *dew point* temperature or 55° F. and is 0.009 pound.

Humidifying. Assuming a room temperature of 70° F. and 40 per cent relative humidity to be maintained when the outside temperature is 0° F. Locate the intersection of the vertical 70° F. dry-bulb temperature with the 40 per cent relative humidity curve, follow the *constant weight vapor line*, passing through the intersection, to its intersection with the saturation curve or 45° F. corresponding to 0.0063 lb. vapor per lb. of dry air. This is then the temperature at which the saturated air must leave the washer, and is the temperature for which the thermostat controlling the spray water heater must be set.

The heat per pound of dry air required for the tempering coil and water heater is read at the right of the diagram, and is 17.5 B.t.u. to raise the temperature of the incoming air from 0° F. to 45° F. and saturate it at this temperature. The additional heat required for the reheater will depend upon the final temperature desired for the air entering the room.

Air Cooling. Entering air 89° F. dry bulb and a relative humidity of 50 per cent corresponding to a wet-bulb temperature of 74° F., wet-bulb depression 89 − 74 or 15 degrees. If the *humidifying efficiency* of a washer is 60 per cent, then the temperature drop will be 15 × 0.60 or 9 degrees. Temperature of leaving air 89 − 9 or 80 degrees. The wet-bulb temperature remains constant.

Drying. Outside air temperature 80° F. and 40 per cent relative humidity. Heater to raise the temperature to 110° F., at which temperature it is introduced into the drier. Air to leave the drier, 70 per cent saturated. From the intersection of the vertical 80° F. dry-bulb temperature line and the 40 per cent relative humidity curve, follow the diagonal *equal weight of vapor line* until it intersects the vertical 110° F. line corresponding to 15 per cent relative humidity. Read horizontally to the left to the intersection with the 70 per cent curve. The corresponding dry-bulb temperature is 81° F., which is the required temperature of the leaving air.

The weight of moisture evaporated per lb. of dry air circulated is the difference between the weight of vapor per lb. of dry air for 81° F. and 70 per cent humidity and 80° F. and 40 per cent humidity or 0.0159 − 0.0088 = 0.0071 lb.

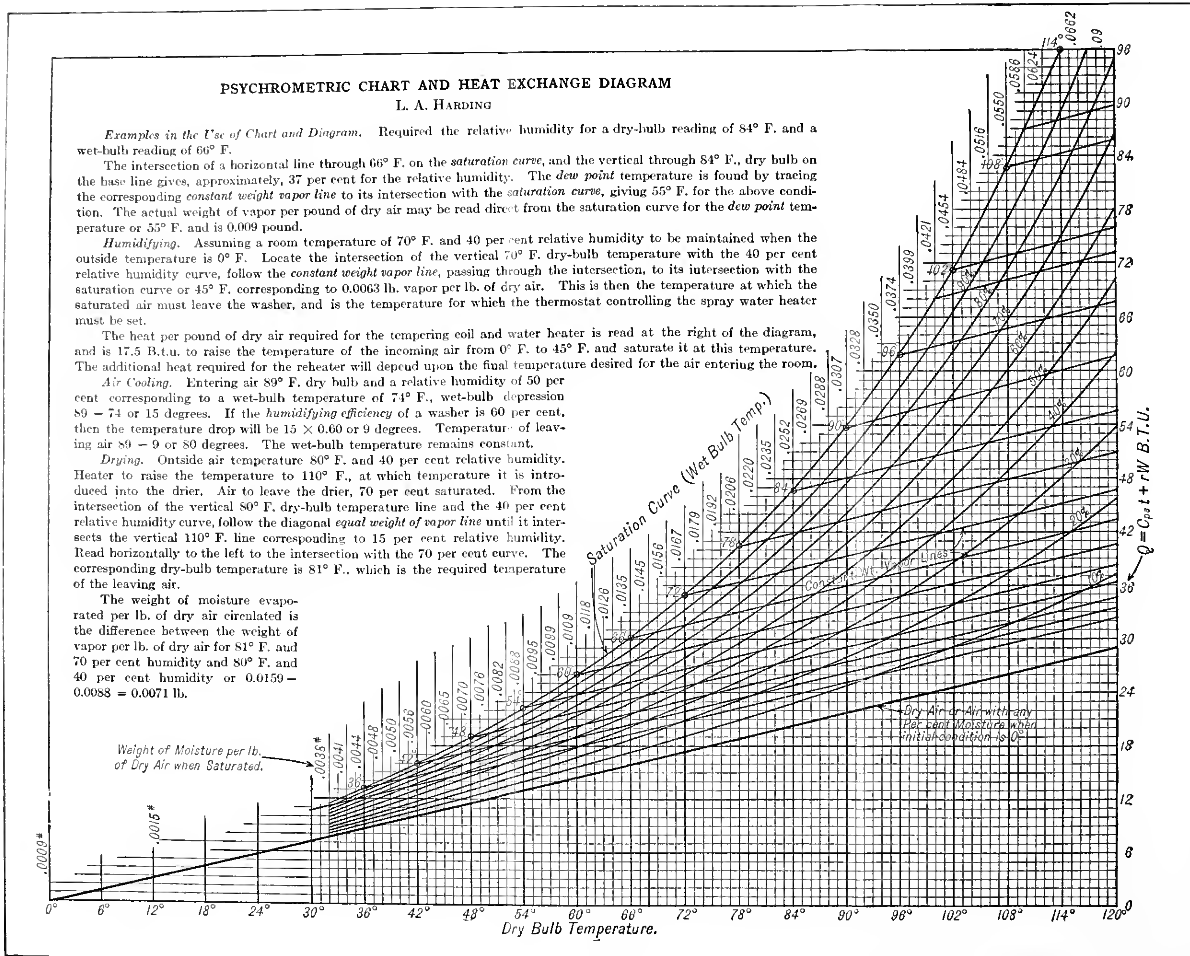


Fig. 2

that a negative correction for radiation of approximately 1.6 per cent of the wet bulb depression should be made to obtain the true depression.

A more refined type of apparatus, known as the *Assmann Aspirating Psychrometer*, makes use of a small fan to draw air over the thermometer bulbs at a constant rate, and in addition each bulb is carefully shielded to protect it from radiation.

Heat Exchange Diagram and Psychrometric Chart. The heat exchange diagram (Fig. 2) is plotted using temperatures as abscissæ and B.t.u. as ordinates. The heat required to raise the temperature of 1 lb. of dry air from 0° to any temperature t is equal to $C_{pa}\,t$ (C_{pa} = specific heat of air at constant pressure = 0.2411). The *dry air* line having been drawn as shown, the *saturation* curve is plotted by adding the heat required, $r\,W$, to evaporate the weight of vapor mixed with saturated air, as may be calculated, to that of one pound of dry air above 0°, for the same temperature. The heat required to raise the temperature of one pound of dry air from zero to the required temperature and evaporate the weight of moisture added to saturate the air is known as the heat content of saturated air and is expressed by the formula $C_{pa}\,t + r\,W$.

To find the per cent relative humidity when the wet bulb reading is 66° F. and dry bulb reading is 84° F. The intersection of the horizontal line through 66° F. on the saturation curve and the vertical through 84° F. dry bulb temperature gives approximately 37 per cent for the relative humidity.

The *dew point* temperature for the above condition is found by following the diagonal *constant weight vapor line* to its intersection with the saturation curve giving 55° F+.

The actual weight of vapor mixed with one pound dry air is therefore 0.37 × 0.0252 (weight of vapor per lb. of dry air when saturated at 84° F.) or 0.009. This may be read direct on the saturation curve for 55° F. The *dew point* temperature should not be confounded with the temperature of adiabatic saturation which is always recorded by the wet bulb thermometer and in this case is 66° F.

AIR WASHING AND HUMIDIFYING

Necessity for Humidifying Air. The practice of *washing the air introduced* for ventilation to remove the dust and other impurities in all cases affects its humidity.

The necessity of some means of artificial humidification for heated buildings during cold dry weather is now quite generally conceded by engineers and physicians who have made a study of the subject. The commercial advantages of artificial humidification in textile mills are well known and appreciated by manufacturers of this class of goods.

Mr. *J. I. Lyle*, discussing the subject of relative humidity in a paper read before the *A. S. H. & V. E. Society*, 1913, draws the following conclusions:

1. That there is little evidence to indicate with any degree of exactness the most desirable humidity for health and comfort, excepting that extremes should be avoided, which seems to be well established, and the limits would seem to be between 30 per cent and 80 per cent.

2. That low humidities are uncomfortable unless accompanied by high temperatures.

3. Low humidities are detrimental to health, by the increase in the amount of dust floating in the atmosphere carrying injurious bacteria, and by irritation of the nervous system through excessive dryness of the skin and excessive evaporation, thus producing nervous tension even in a healthy person.

4. With single sash in cold weather high humidity will be objectionable on account of excessive condensation taking place and running from the windows over the woodwork and wall decorations.

5. In order best to satisfy the second, third and fourth conditions in cold weather, the relative humidity should be maintained between 35 per cent and 45 per cent.

The necessity for artificial humidification during cold weather in buildings will be made apparent by the following illustration:

Example. Assume that a temperature of 70° F. is maintained in a building artificially heated when the outside air temperature is 0° F. and its relative humidity is 50 per cent. Required, the relative humidity that will exist in the building if the air is not humidified. Each pound of entering dry air will have mixed with it 0.0008 × 0.50 or 0.0004 lb. of vapor. When heated to a temperature of 70° F. it still contains the same weight of vapor per pound. Saturated air at 70° F. contains 0.0156 lb. of vapor per lb. of dry air. The relative humidity will therefore be 0.0004/0.0156 or 2.6 per cent. This creates an artificial climate as dry as that of any desert and demonstrates the necessity for artificial humidification.

Humidifying the Air. Assume (Fig. 4) that a constant temperature of 70° F. and 40 per cent relative humidity is desired when the outside air temperature is 0° F. and the relative humidity is 25 per cent.

The incoming air is first passed through a tempering coil which raises the temperature of the incoming air to about 40° F. A tempering coil made up of two sections, 8 rows of pipe, with an air velocity of 1200 ft. per min. through the *free area* of coil, and a steam pressure of 5 lb. gage will accomplish this result. The object of the tempering coil is to prevent freezing in the spray chamber in extremely cold weather, and relieve the washer of part of the work necessary in supplying the heat to raise the temperature of the incoming air.

The air in passing through the warmed spray water, which is heated by means of a steam injector type of heater, will have its temperature raised and will leave the spray chamber in a saturated condition. *This temperature is fixed by the relative humidity desired in the room,* as the weight of moisture per pound of dry air must be the same here as when entering the room. In this particular example the requirements are 70° F. and 40 per cent relative humidity (wet bulb 58° F.), corresponding to 0.0156 × 0.40 or 0.00624 lb. moisture per lb. of dry air. The saturation temperature corresponding to this weight of moisture is 45° F., which will be the wet and dry bulb temperature of the air leaving the spray chamber.

This temperature must evidently remain a constant regardless of the temperature and relative humidity of the entering air.

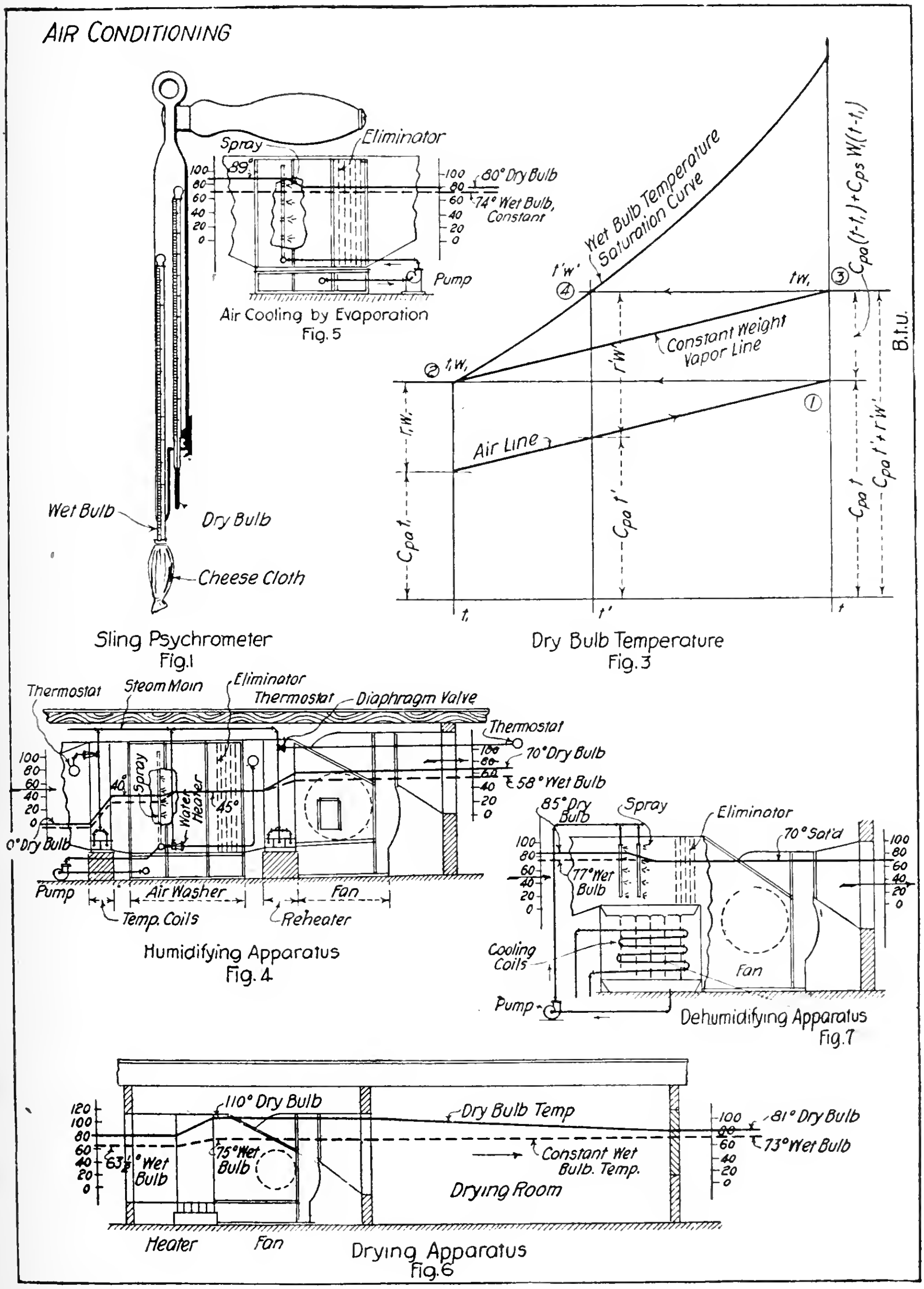
The *heat exchange* taking place in the spray chamber between the air and water is as follows: The air brought into contact with the spray water, maintained at a higher temperature, will first become saturated at a temperature corresponding to the temperature of the air entering the spray chamber or 40° F. in this case. If the water is still sufficiently warm the evaporation will continue as the temperature is raised, the air continuing to be saturated, to the predetermined temperature or, in this example, 45° F.

The *thermostat at the washer outlet* is set for 45° F., so that more or less steam is turned into the water heater, as may be required to raise the temperature of the air and furnish sufficient heat to evaporate the amount of moisture necessary to saturate the air at 45° F. The thermostat referred to operates on a diaphragm, through the medium of compressed air, controlling the steam admission valve on the ejector water heater. The air leaving the washer is then passed through the reheater, where its temperature is raised to the final requirement of 70° F. and a relative humidity of 40 per cent.

The temperature of the reheating coils is controlled by the thermostat, located in the room, which operates the diaphragm valve on the steam supply through the medium of compressed air.

It is evident from the foregoing, that humidity control is obtained by maintaining a *constant saturation temperature of the air leaving the washer* by controlling the amount of heat supplied the spray water. See chapter on "Temperature and Humidity Control."

Heat Required to Condition Air. The heat, per pound of dry air, required for the tempering coil and water heater is equal to the sum of the heats required to raise the temperature of the incoming air to the saturation temperature of the air leaving the washer and the latent heat



Sling Psychrometer
Fig. 1

Air Cooling by Evaporation
Fig. 5

Dry Bulb Temperature
Fig. 3

Humidifying Apparatus
Fig. 4

Dehumidifying Apparatus
Fig. 7

Drying Apparatus
Fig. 6

to evaporate the weight of moisture added to saturate the air at this temperature. Thus for the above example we have

$$C_{pa}\, t + r'W' = 0.24 \times 45 + 1{,}066 \times 0.0063 = 17.51 \text{ B.t.u.}$$

This may be read on the right-hand scale of the chart on a horizontal line passing through 45° on the saturation curve.

For any other initial temperature of above 0 subtract the heat content for this condition from the heat content for the final condition. Thus, if the entering air temperature is 36° and 40 per cent relative humidity the heat content for this condition is 10 B.t.u. The net heat to be supplied is therefore 17.51 − 10 or 7.51 B.t.u. per pound of dry air.

The additional heat required for the reheater will depend upon the final temperature desired for the air entering the rooms.

Air Cooling by Evaporation. When air is passed through an air washer or humidifier and the *spray water is simply recirculated and not heated* a constant interchange of heat takes place between the air and water. A portion of the water is evaporated by the heat supplied from the air in having its temperature lowered.

The dry-bulb temperature is lowered but *the wet-bulb temperature remains constant.*

The *ultimate temperature* that could be obtained would be the temperature of adiabatic saturation which is the wet-bulb temperature of the entering air.

The following example, will serve to illustrate the principles involved:

Example. The incoming air is assumed to have a dry-bulb temperature of 89° F., and a wet-bulb temperature of 74° F. +, corresponding to 50 per cent relative humidity and carrying 0.015 lb. of vapor or moisture per lb. of dry air (Fig. 5). The weights of moisture for the conditions given may be read directly from the chart. The air leaving the washer is assumed to have been lowered in temperature to 80° F. dry bulb, the wet-bulb temperature remaining constant or 74° F. +. Each pound of leaving air will now contain 0.017 lb. of vapor, the washer or humidifier having added the difference 0.017 − 0.015 or 0.002 lb. of vapor or moisture per lb. of air circulated. This is an adiabatic change, the heat given up by the air and its contained vapor in lowering its temperature from 89° F. to 80° F. or 9° goes to evaporate the weight of moisture added. The weight added $(W' - W)$ may be calculated by the equation, $r'(W' - W) = C_{pa}(t - t') + C_{ps} W (t - t')$. Heat required to evaporate vapor added = heat given up by the air and its vapor

$$W' - W = \frac{0.24\,(89 - 80) + 0.44 \times 0.015 \times (89 - 80)}{1{,}049.8} = 0.0021 \text{ lb. per lb. of air.}$$

The latent heat r' is that corresponding to the wet-bulb temperature (74° F. +) of the incoming air or 1,049.8 B.t.u.

Humidifying Efficiency. If it were possible for the washer to fully saturate the air it would leave with the same wet- and dry-bulb temperature, or 74° F. +, giving a cooling effect of 89 − 74 or 15°. The ratio of cooling done by this particular washer, termed *humidifying efficiency* by *Mr. J. I. Lyle* (*Trans. A. S. H. & V. E., Vol.* 18) is therefore $\frac{9}{15}$ or 0.60. *Mr. Lyle* also states: "The percentage would hold good for this same washer regardless of the difference between the wet- and dry-bulb temperature of the entering air.

"The various commercial washers on the market vary in humidifying efficiency from 40 to 75 per cent and a fair average may be taken as 60 per cent. The factors controlling the humidifying efficiency of any washer are:

1. The fineness with which the spray is atomized.
2. The length of the spray chamber.
3. The impact with which the air and water meet and are mixed (depends on velocity of air through washer and water pressure at the spray nozzles).
4. The relative quantities or weight of air and water used."

Dehumidifying the Air. If the *temperature of the spray water is maintained below the dew point temperature of the entering air,* the temperature of the entering air and its contained vapor will be first lowered to the dew point temperature, then some of the vapor will be condensed out by the removal of its latent heat, until the final temperature of the leaving saturated air is reached.

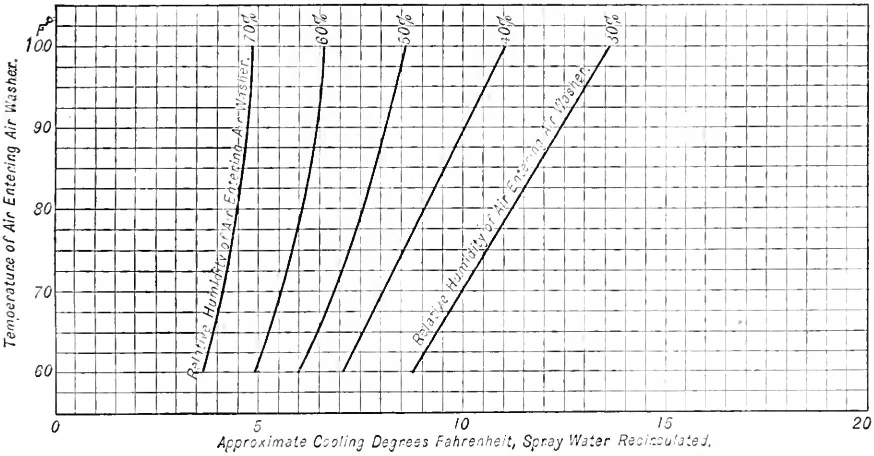


FIG. 8. COOLING CHART FOR WEBSTER STANDARD AIR WASHER.

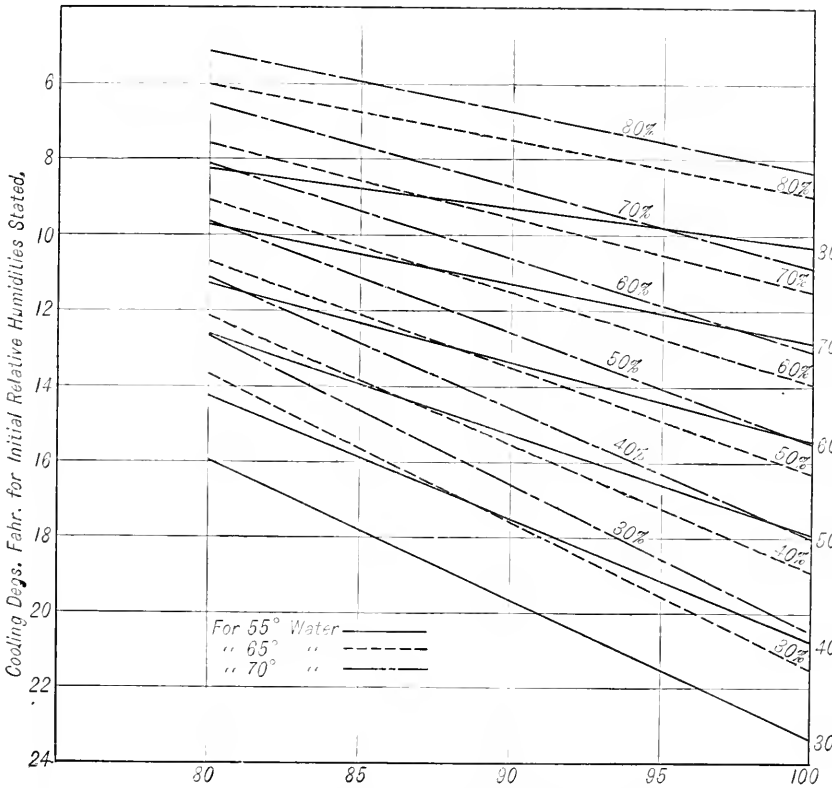


FIG. 9. COOLING CHART—TYPE A WEBSTER WASHER.

4.18 gal. water per 1000 cu. ft. air per min. Average velocity, air through washer, 433 ft. per min. Water pressure, 20 lb. per sq. in. Curves for an initial temperature water of 55°-65°-70°.

In a properly designed dehumidifying apparatus the temperature of the leaving saturated air is within about 1 degree of the temperature of the leaving water.

Heat to be removed. The heat to be extracted from the air and its contained moisture may be determined very closely by the following approximate method.

Let t_1 = initial temperature of the entering air (dry bulb).
t_d = dew point temperature of the air.
t_2 = final temperature of the air.
W_1 = weight of moisture per lb. of air initial condition.
W_2 = weight of moisture per lb. of air final condition.
$W_1 - W_2$ = weight of vapor condensed and removed by apparatus.
r = average latent heat corresponding to temperature t_d and t_2. This is an approximation, but close enough for practical purposes.

$C_{ps}\ (t_1 - t_d)\ W_1 + C_{pa}\ (t_1 - t_d)$ = heat extracted to lower temperature of the air and its contained moisture to the dew point temperature.

Neglecting the small amount of heat extracted to lower the temperature of the weight of moisture condensed out we have

$C_{ps}\ (t_d - t_2)\ W_1 + C_{pa}\ (t_d - t_2)$ as the heat to be extracted to lower the air and its contained moisture from the dew point temperature (t_d) to the final saturation temperature (t_2).

The total heat to be extracted per lb. of entering dry air is therefore:

$$r\ (W_1 - W_2) + C_{ps}\ (t_1 - t_2)\ W_1 + C_{pa}\ (t_1 - t_2) \text{ B.t.u.} \qquad (1)$$

The amount of heat to be extracted per lb. of air may be read direct from the chart Fig. 2, by taking the difference between the heat required to saturate air at the dew point temperature t_d and raise it to temperature t_1, and the heat required to saturate air at the final saturation temperature t_2.

If t_x = initial temperature of spray water.
t_y = final temperature of spray water.
S = weight of water circulated per lb. of dry air passing through apparatus.
$S\ (t_x - t_y)$ = B.t.u. given up by the water
= the total heat to be extracted as given above.

$$\text{Then} \quad S = \frac{\text{B.t.u. to be extracted per lb. of air}}{t_x - t_y}.$$

The cold spray water used to lower the temperature of the entering air and condense out the moisture is either taken from an artesian well (the temperature of artesian well water is usually assumed about 55° F.) or may be water previously cooled by artificial refrigeration.

When the water is artificially cooled the refrigerating coils, either direct expansion or brine coils are placed underneath the spray chamber and the water so distributed as to flow uniformly over the coils and collect in the tank beneath. A centrifugal pump is used to circulate the water (Fig. 7).

If the spray water is simply recirculated without cooling it, the spray water temperature will be reduced to approximately 60 per cent of the wet bulb depression and is the limit of the cooling effect under this condition of operation as was previously shown under "Air Cooling by Evaporation."

Example. Required the capacity of a refrigerating machine in tons of refrigeration per 24 hours, (1 ton = 288,000 B.t.u.) to dehumidify 20,000 cu. ft. of air per minute. Entering air 85° F. and 70 per cent humidity, air to be cooled down to 70° F. (Fig. 7).

$t_1 = 85°$ F., $t_2 = 70°$ F., $W_1 = 0.026 \times 0.70 = 0.0182$. This weight corresponds to a dew point temperature t_d of 75° F. (nearly) $W_2 = 0.0156$, $r = 1{,}049$. Substituting the above values in equation (1), we have, $1{,}049 \times (0.0182 - 0.0156) + 0.44 \times (85 - 70) \times 0.0182 + 0.24 \times (85 - 70) = 6.45$ B.t.u. to be

extracted per lb. of dry air. The weight of dry air circulated per minute is 20,000 × 0.072 (density at 85° F.) = 1,440 lb. Then, $\frac{1,440 \times 6.45 \times 60 \times 24}{288,000}$ or 46.5 tons of refrigeration per 24 hours are required.

The heat to be extracted as determined by means of the psychrometric chart is found as follows. The heat for 85° F. and 70 per cent relative humidity as read is 39.5 B.t.u. and for 70° F. and saturated is 33.2 B.t.u. giving a difference of 39.5 − 33.2 or 6.3 B.t.u. per lb. of air.

The temperature of the water leaving will be approximately 1° lower than the leaving air or 69° F. Assuming an initial water temperature of 59° F. then $\frac{1,140 \times 6.45}{69 - 50}$ = 736.4 lb. or 98 gals. of water per minute to be circulated. The temperature of the expanding ammonia if direct expansion coils are used will be approximately 20° lower or 39° F., corresponding to a suction or back pressure of 58 lb. per sq. in. gage.

Power Required for Operating Humidifiers. The *power required per* 1,000 *cu. ft. of air per minute* for the apparatus may be calculated, approximately, by assuming 4.5 gallons of water circulated at a pressure of 25 lb. per sq. in. or 60 ft. head and a centrifugal pump efficiency of 0.40.

The pump horsepower is therefore equal to, $\frac{4.5 \times 8.33 \times 60}{33,000 \times 0.40}$ or 0.171.

The fan horsepower depends upon the resistance of the ducts, heater and washer and the fan efficiency. Assuming a total resistance of one inch of water and a fan efficiency of 50 per cent, the fan brake horsepower will be $\frac{5.2 \times 1 \times 1,000}{33,000 \times 0.50}$ or 0.315.

For the assumed conditions the total power required will therefore be 0.171 + 0.315 or 0.486 horsepower per 1000 cu. ft. of air per minute. The electrical horsepower input to the motors will be approximately 1½ times this amount when the efficiency of the motors is taken into account.

The table following, by *W. H. Carrier, Trans. A. S. M. E.*, Dec., 1911, exhibits the power required to saturate 1000 cu. ft. of air per minute at various velocities. This is based on overcoming the resistance of the humidifier, using a fan with a static efficiency of 45 per cent, a fair value:

TABLE 2

RESISTANCE OF CARRIER HUMIDIFIERS AND HORSEPOWER REQUIRED TO HUMIDIFY 1000 CU. FT. OF AIR

Velocity Through Spray Chamber in Ft. per Min.	Resistance in In. of Water	Horsepower to Move 1000 Cu. Ft. Air per Min. At 45 per Cent Fan Efficiency	Horsepower for Spray per 1000 Cu. Ft. of Air (1/16" Orifice Nozzle)	*Total Horsepower Required per 1000 Cu. Ft. of Air
350	0.112	0.0391	0.1408	0.1799
400	.147	.0513	.1231	.1744
450	.186	.0652	.1095	.1747
500	.229	.0800	.0985	.1785
550	.277	.0968	.0897	.1865
600	.330	.1150	.0822	.1972
650	.387	.1350	.0758	.2108
700	.450	.1570	.0704	.2274
750	.516	.1810	.0658	.2468

* NOTE.—This does not include the power required to overcome the resistance of the ducts, which varies considerably, but which should not exceed that required for the humidifier. The resistance of the heating coils is not considered, because in summer when the largest supply of air is usually required the air is by-passed around the heaters, while in winter the requirements are so much smaller that the total horsepower is greatly reduced and the total resistance is but slightly increased.

The power required to pump the water is based on the use of centrifugal pumps giving 55 per cent efficiency and using 1/16-in. orifice nozzles with rotary self-cleaning strainers.

Standard Types of Air Conditioning Apparatus. Standard types of air washers are now obtainable for the following classes of service:

(1) *Standard Type Air Washer.* This apparatus is designed primarily for air cleansing, in connection with heating and ventilating apparatus where maximum cooling in summer is relatively unimportant, as for public and semi-public buildings. The maximum cooling effect obtained is approximately 60 per cent of the wet-bulb depression of the entering air. One bank of spray or mist nozzles only is used with this type.

Provision is made for heating the spray water and controlling the moisture content of the air in cold weather.

(2) *Air Washer and Humidifier.* This apparatus is designed for use where maximum cooling of the air by evaporation in addition to air cleansing is desired with recirculated spray water.

The maximum cooling effect obtained being approximately 85 per cent of the wet-bulb depression of the entering air, this type is used principally in manufacturing plants in which the requirements covering the conditioning of the air cannot be met by the standard type washer.

Provision is made for heating the spray water as in (1), so that the moisture content may be controlled in cold weather.

Complete saturation of the air is possible with this type when the spray water is heated.

(3) *Air Washer and Dehumidifier.* This type of apparatus is designed primarily for cooling or dehumidifying the air by the use of spray water cooled by mechanical refrigeration or as may be supplied from artesian wells. Dehumidifying apparatus is frequently an essential feature, necessary in certain processes of manufacture, during summer weather conditions when the relative humidity is high and the moisture content of the air after passing through a standard washer or humidifier would be too high for the process involved.

Complete saturation of the air is possible with this type of apparatus.

Rating of Air Washers. Air washers are designed to handle air at the rate of 400 to 500 cu. ft. per min. per sq. ft. of cross-sectional area through the spray chamber.

The pump and nozzles must handle between 4 and 5 gal. of water per 1000 cu. ft. of air per min. The total head against which the pump operates is from 45 to 50 ft. The efficiency of the smaller size centrifugal pumps is rather low, as will be noted under the chapter on "Pumps", and may be assumed as 40 per cent.

Spray nozzles are designed to handle approximately $1\frac{1}{4}$ gal per min.

The number of nozzles required for a given installation will be approximately:

$$N = \frac{\text{cu. ft. per min. air capacity of washer} \times 4.5}{1000 \times 1.25}$$

The capacity of the circulating pump required is $C = \dfrac{\text{cu. ft. per min.}}{1000 \times 1.25}$ gal. per min.

TABLE 3

FRICTION PRESSURE LOSS THROUGH WEBSTER AIR WASHER

Velocity Feet per Minute	Pressure Loss in In. of Water
100	0.01
200	.04
300	.09
400	.16
500	.25
600	.36
700	.49
800	.64

TABLE 4

TEMPERATURE TO WHICH AIR CAN BE COOLED WITH AIR WASHER

(Spray Engineering Co.)

Temp. of Air Entering Washer Degrees Fahr.	Relative Humidity per Cent 30.0 Inch Barometer													
	25	30	35	40	45	50	55	60	65	70	75	80	85	90
	Temperature of Air Leaving Washer with Cooling Water Recirculated													
50........	37.5	38.5	39.5	40.4	41.4	42.1	43.0	43.9	44.1	45.4	46.1	47.0	47.8	48.5
55........	41.4	42.4	43.4	44.4	45.3	46.4	47.1	48.1	49.0	50.0	50.9	51.1	52.5	53.4
60........	44.9	46.0	46.9	48.4	49.4	51.0	51.5	52.5	53.5	54.5	55.5	56.4	57.4	58.4
65........	48.5	49.8	51.0	52.3	53.3	54.5	55.8	56.8	57.3	59.0	60.0	61.0	62.0	63.0
70........	52.0	53.3	54.8	56.0	57.3	58.8	60.0	61.3	62.3	63.5	64.6	65.8	66.8	68.0
75........	55.3	57.0	58.5	60.0	61.5	62.8	64.3	65.5	66.8	68.0	69.3	70.5	71.8	72.8
80........	58.8	60.5	62.0	63.8	65.5	67.0	68.5	69.8	71.3	72.5	74.0	75.3	76.5	77.8
85........	62.2	64.0	65.8	67.7	69.5	71.0	72.5	74.1	75.6	77.3	78.6	80.0	81.2	82.4
90........	65.5	67.5	69.5	71.5	73.5	75.0	76.5	87.4	80.0	82.0	83.2	84.8	86.0	87.2
95........	69.0	71.3	73.3	75.5	77.5	79.3	81.0	82.8	84.5	86.5	87.9	89.3	90.8	92.2
100........	72.5	75.0	77.0	79.5	81.5	83.5	85.5	87.2	89.0	91.0	92.5	94.0	95.5	97.2
105........	75.8	78.5	80.8	82.3	85.5	87.8	89.8	91.6	93.5	95.5	97.3	98.7	100.4	102.1
110........	79.0	82.0	84.5	87.0	89.5	92.0	94.0	96.0	98.0	100.0	102.0	103.5	105.2	107.0
115........	82.5	85.8	88.6	91.0	93.8	96.3	98.5	100.5	102.5	101.6	106.6	108.2	110.1	111.9
120........	86.0	89.5	92.5	95.0	98.0	100.5	103.0	105.0	107.0	109.2	111.2	113.0	115.0	116.8

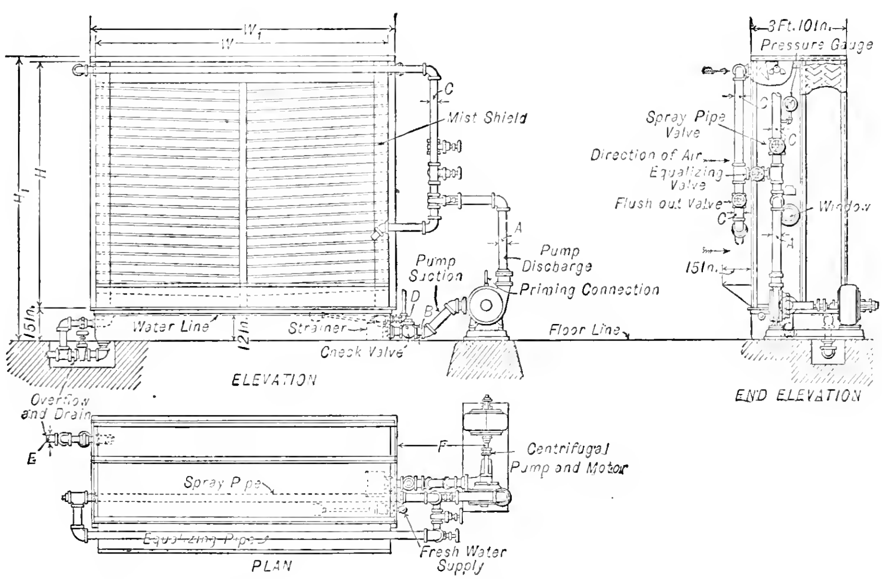


FIG. 10. WEBSTER STANDARD AIR WASHER WITH SHEET METAL TANK.

TABLE 5

DIMENSIONS OF WEBSTER STANDARD AIR WASHERS

(See Fig. 10)

Capacity	H	W	H_1 With		W_1	Capacity	H	5/8	H_1 With		W_1
			Metal Tank	Concrete Tank					Metal Tank	Concrete Tank	
1,000	1′6″	1′6″	2′10 ½″	1′ 7½″	1′ 9″	40,000	9′8″	8′3″	11′ ½″	9′ 9½″	8′ 6″
2,000	2′9″	1′6″	4′ 1½″	2′10½″	1′ 9″	45,000	9′8″	9′4″	11′ ½″	9′ 9½″	9′ 7″
3,000	4′0″	1′6″	5′ 4½″	4′ 1½″	1′ 9″	50,000	9′8″	10′4″	11′ 1″	9′10″	10′ 8″
4,000	4′0″	2′0″	5′ 4½″	4′ 1½″	2′ 3″	55,000	9′8″	11′4″	11′ 1″	9′10″	11′ 8″
5,000	4′0″	2′6″	5′ 4½″	4′ 1½″	2′ 9″	60,000	9′8″	12′4″	11′ 1″	9′10″	12′ 8″
7,500	5′0″	3′0″	6′ 4½″	5′ 1½″	3′ 3″	65,000	9′8″	13′6″	11′ 1″	9′10″	13′10″
10,000	5′9″	3′6″	7′ 1½″	5′10½″	3′ 9″	70,000	9′8″	14′6″	11′ 1″	9′10″	14′10″
12,500	7′0″	3′6″	8′ 4½″	7′ 1½″	3′ 9″	75,000	9′8″	15′6″	11′ 1″	9′10″	15′10″
15,000	7′6″	4′0″	8′10½″	7′ 7½″	4′ 3″	80,000	9′8″	16′6″	11′ 1″	9′10″	16′10″
20,000	8′0″	5′0″	9′ 4½″	8′ 1½″	5′ 3″	85,000	9′8″	17′6″	11′ 1″	9′10″	17′10′
25,000	9′0″	5′6″	10′ 4½″	9′ 1½″	5′ 9″	90,000	9′8″	18′8″	11′ 1″	9′10″	19′ 0″
30,000	9′8″	6′3″	11′ ½″	9′ 9½″	6′ 6″	95,000	9′8″	19′8″	11′ 1″	9′10″	20′ 0″
35,000	9′8″	7′3″	11′ ½″	9′ 9½″	7′ 6″	100,000	9′8″	20′8″	11′ 1″	9′10″	21′ 0″

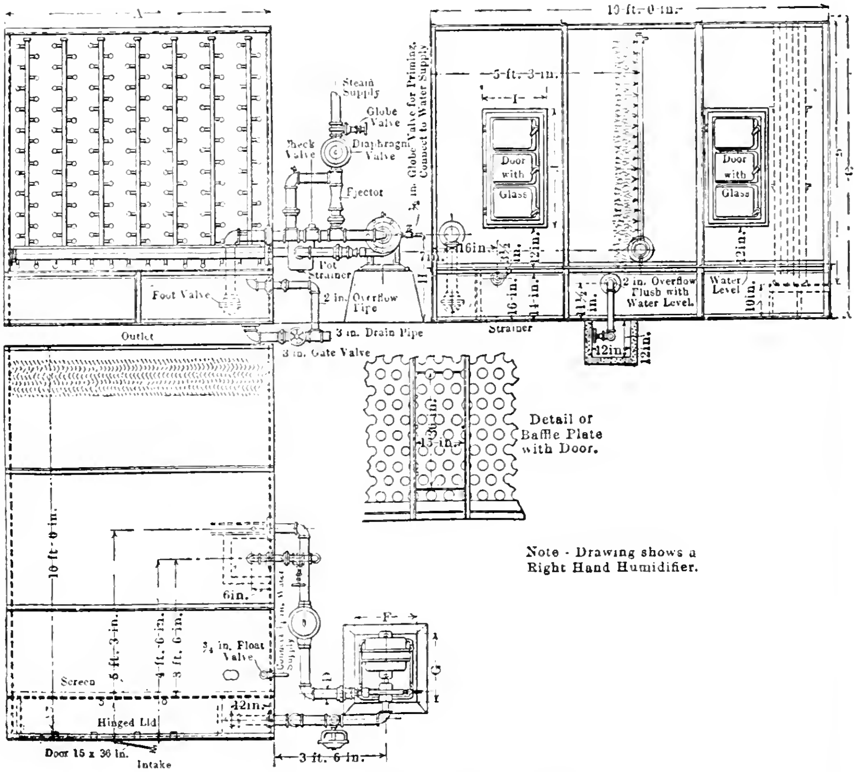


FIG. 11. CARRIER STANDARD HUMIDIFIER.

TABLE 6

DIMENSIONS FOR CARRIER STANDARD HUMIDIFIERS WITH HUMIDITY CONTROL CONNECTIONS

(See Fig. 11)

Number	Cubic Feet Air per Minute	A	B	C	D	E	F	G	H	I	J
		Ft. Ins.	Feet	Ft. Ins.	Ins.	Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.	Ft. Ins.
51—A	4,025	2— 2	3	3—11½	1½	1½	2—2	2—3	2—1	1—6¾	2—1¾
51—B	5,481	2— 2	4	4—11½	1½	1½	2—2	2—3	2—1	1—6¾	2—1¾
51—C	8,050	2— 2	6	6—11½	1½	1½	2—2	2—3	2—1	1—6¾	3—4
52—B	8,050	3— 1½	4	4—11½	1½	1½	2—2	2—3	2—1	1—6¾	2—1¾
52—C	12,060	3— 1½	6	6—11½	2	2	2—3½	2—4½	2—13/16	1—6¾	3—4
52—D	16,100	3— 1½	8	8—11½	2	2	2—3½	2—4½	2—13/16	1—6¾	3—4
53—C	16,100	4— 1	6	6—11½	2	2	2—3½	2—4½	2—13/16	1—6¾	3—4
53—D	21,462	4— 1	8	8—11½	2	2	2—3½	2—4½	2—13/16	1—6¾	3—4
53—E	26,831	4— 1	10	10—11½	2½	2½	2—4	2—6	2—1⅛	1—6¾	3—4
54—C	20,000	5— ½	6	6—11½	2	2	2—3½	2—4½	2—13/16	1—6¾	3—4
54—D	26,600	5— ½	8	8—11½	2½	2½	2—4	2—6	2—2⅛	1—6¾	3—4
54—E	33,250	5— ½	10	10—11½	2½	2½	2—4	2—6	2—2⅛	1—6¾	3—4
55—D	32,200	6— 0	8	8—11½	2½	2½	2—4	2—6	2—2⅛	1—6¾	3—4
55—E	40,250	6— 0	10	10—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
56—D	37,600	6—11½	8	8—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
56—E	46,970	6—11½	10	10—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
57—D	42,931	7—11	8	8—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
57—E	53,662	7—11	10	10—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
58—D	48,300	8—10½	8	8—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
58—E	60,400	8—10½	10	10—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
59—D	53,620	9—10	8	8—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
59—E	67,100	9—10	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
60—D	59,500	10— 9½	8	8—11½	2½	3	2—4½	2—8	2—5	1—6¾	3—4
60—E	73,780	10— 9½	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
61—D	64,000	11— 9	8	8—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
61—E	79,800	11— 9	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
62—D	69,720	12— 8½	8	8—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
62—E	87,150	12— 8½	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
63—D	75,110	13— 8	8	8—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
63—E	93,870	13— 8	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
64—E	100,660	14— 7½	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
65—E	107,310	15— 7	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
66—E	114,100	16— 6½	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
67—E	120,750	17— 6	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4
68—E	127,470	18— 5½	10	10—11½	3½	4	2—5¼	2—9½	2—7	1—6¾	3—4

TABLE 7

CAPACITY OF SIZE "A," TYPE "A," AIR PURIFIERS

(See Fig. 12)

Number of Purifier		A-1	A-2	A-3	A-4	A-5	A-6	A-7
C. F. M.		5,700	8,000	10,000	12,500	14,500	17,000	19,000
Gallons per minute		24	36	48	60	72	84	96
Total head		45	45	45	45	45	45	45
Size of pump		1¼	1¼	1¼	1½	1½	1½	2
Size of suction		1½	1½	1½	2	2	2	2½
H. P. of motor		2	2	2	3	3	3	3
Fresh water connection		¾	¾	¾	¾	¾	¾	¾
Drain connection		2	2	2	2	2	2	2
Area, sq. ft.		11.4	16	20	25	29	34	38
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all dimensions	A	4′ 8″	4′ 8″	4′ 8″	4′ 8″	4′ 8″	4′ 8″	4′ 8″
	B	4′ 1″	5′ 7″	7′ 1″	8′ 7″	10′ 1″	11′ 7″	13′ 1″
	C	9″	9″	9″	10″	10″	10″	11″
	D	37″	37″	37″	44″	44″	44″	44″
	E	34″	34″	36″	36″	36″	37″	37″
	F	3′10″	5′ 4″	6′10″	8′ 4″	9′10″	11′ 4″	12′10″
	G	3′ 0″	3′ 0″	3′ 0″	3′ 0″	3′ 0″	3′ 0″	3′ 0″
	H	14″	14″	14″	14″	14″	14″	14″

TABLE 7—*Continued*

CAPACITY OF SIZE "B," TYPE "A," AIR PURIFIERS

Number of Purifier		B-1	B-2	B-3	B-4	B-5	B-6	B-7	B-8
C. F. M.		8,500	12,000	15,000	19,000	22,000	25,000	29,000	32,000
Gallons per minute		36	54	72	90	108	126	144	162
Total head		47	47	47	47	47	47	47	47
Size of pump		1¼	1½	1½	2	2	2	2½	2½
Size of suction		1½	2	2	2½	2½	2½	3	3
H. P. of motor		2	3	5	5	5	5	5	5
Fresh water connection		¾″	¾″	¾″	¾″	¾″	¾″	¾″	¾″
Drain connection		2	2	2	2	2	2	2	2
Area, sq. ft.		17	26	30	38	44	50	58	64
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all dimensions	A	6′ 2″	6′ 2″	6′ 2″	6′ 2″	6′ 2″	6′ 2″	6′ 2″	6′ 2″
	B	4′ 1″	5′ 7″	7′ 1″	8′ 7″	10′ 1″	11′ 7″	13′ 1″	14′ 7″
	C	9″	10″	10″	11″	11″	11″	12″	12″
	D	37″	44″	44″	46″	46″	46″	47½″	47½″
	E	34″	36″	36″	37″	37″	37″	40″	40″
	F	3′10″	5′ 4″	6′10″	8′ 4″	9′10″	11′ 4″	12′10″	14′ 4″
	G	4′ 6″	4′ 6″	4′ 6″	4′ 6″	4′ 6″	4′ 6″	4′ 6″	4′ 6″
	H	14″	14″	14″	14″	14″	14″	14″	14″

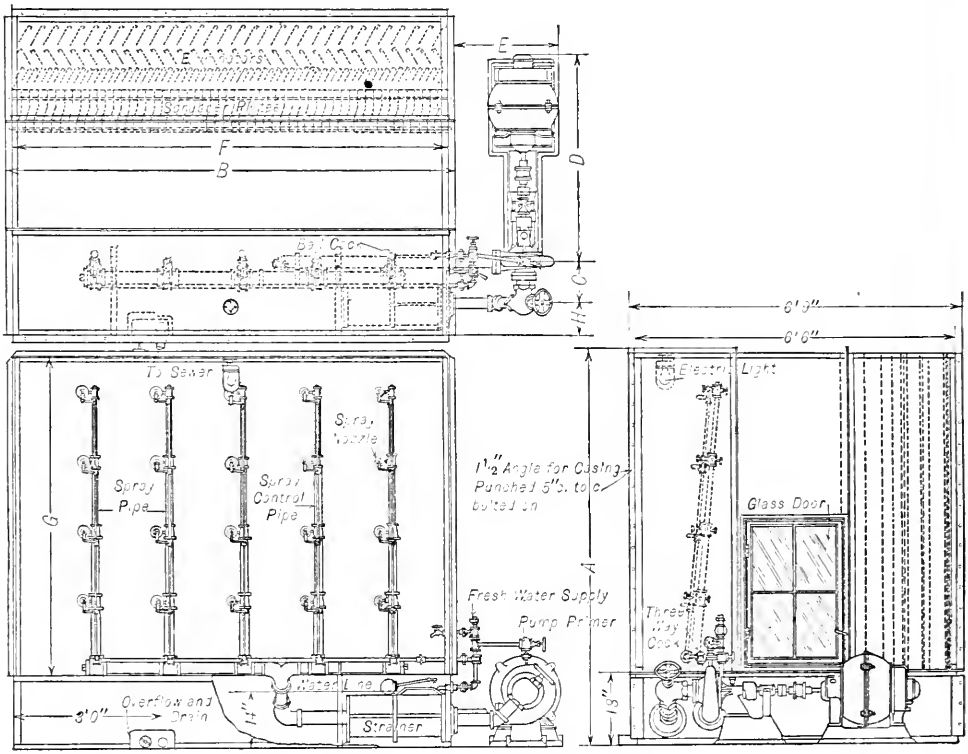


Fig. 12. Sirocco Type "A" Purifier.

TABLE 7—*Continued*

CAPACITY OF SIZE "C," TYPE "A," AIR PURIFIERS

Number of Purifier		C-1	C-2	C-3	C-4	C-5	C-6	C-7	C-8	C-9	C-10	C-11
C. F. M.		14,000	19,000	23,500	28,500	33,000	38,500	43,000	48,000	52,500	58,000	62,500
Gallons per minute		48	72	96	120	144	168	192	216	240	264	288
Total head		50	50	50	50	50	50	50	50	50	50	50
Size of pump		1¼	1½	2	2	2½	2½	2½	2½	2½	3	3
Size of suction		1½	2	2½	2½	3	3	3	3	3	4	4
H. P. of motor		3	3	5	5	5	5	7½	7½	7½	7½	7½
Fresh water connection		¾	¾	¾	¾	¾	¾	¾	¾	¾	1	1
Drain connection		2	2	2	2	2	2	2	2	2	3	3
Area, sq. ft.		28	38	47	57	66	77	86	96	105	116	127
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all dimensions	A	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″	8′ 2″
	B	4′ 7″	6′ 1″	7′ 7″	9′ 1″	10′ 7″	12′ 1″	13′ 7″	15′ 1″	16′ 7″	18′ 1″	19′ 7″
	C	9″	10″	11″	11″	12″	12″	12″	12″	12″	14″	14″
	D	37″	44″	46″	46″	47½″	47½″	47½″	47½″	47½″	51¾″	51¾″
	E	34″	36″	37″	37″	39″	39″	39″	39″	39″	41″	41″
	F	4′ 4″	5′10″	7′ 4″	8′10″	10′ 4″	11′10″	13′ 4″	14′10″	16′ 4″	17′10″	19′ 4″
	G	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″	6′ 6″
	H	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″

TABLE 8

CAPACITY OF SIZE "D," TYPE "A," AIR PURIFIERS

Number of Purifier		D-1	D-2	D-3	D-4	D-5	D-6	D-7	D-8	D-9	D-10	D-11	D-12
C. F. M.		16,000	22,000	27,000	33,000	38,000	44,000	50,000	56,000	61,000	67,000	72,000	78,000
Gallons per minute		60	90	120	150	180	210	240	270	300	330	360	390
Total head		52	52	52	52	52	52	52	52	52	52	52	52
Size of pump		1½	2	2	2½	2½	2½	2½	3	3	3	3½	3½
Size of suction		2	2½	2½	3	3	3	3	4	4	4	5	5
H. P. of motor		3	5	5	5	7½	7½	7½	7½	10	10	10	15
Fresh water connection		¾	¾	¾	¾	¾	¾	¾	1	1	1	1	1
Drain connection		2	2	2	2	2	2	2	2	3	3	3	3
Area, sq. ft.		32	44	54	66	76	88	100	112	122	134	144	156
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all dimensions	A	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″	9′ 2″
	B	4′ 7″	6′ 1″	7′ 7″	9′ 1″	10′ 7″	12′ 1″	13′ 7″	15′ 1″	16′ 7″	18′ 1″	19′ 7″	21′ 1″
	C	10″	11″	11″	12″	12″	12″	12″	14″	14″	14″	16″	16″
	D	44″	46″	46″	47½″	47½″	47½″	47½″	51¾″	51¾″	51¾″	55¾″	55¾″
	E	36″	37″	37″	37″	39″	39″	39″	41″	41″	41″	41″	41″
	F	4′ 4″	5′10″	7′ 4″	8′10″	10, 4″	11′10″	13′ 4″	14′10″	16′ 4″	17′10″	19′ 4″	20′10″
	G	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″
	H	14″	14″	14″	14″	14″	14″	14″	14″	11″	14″	14″	14″

TABLE 8—*Continued*

CAPACITY OF SIZE "E," TYPE "A," AIR PURIFIERS

Number of Purifier		E-2	E-3	E-4	E-5	E-6	E-7	E-8	E-9	E-10	E-11	E-12	E-13
C. F. M.		28,000	35,000	42,000	49,000	56,000	63,000	70,000	77,000	85,000	92,000	99,000	106,000
Gallons per minute		126	168	210	252	294	336	378	420	462	504	516	588
Total head		55	55	55	55	55	55	55	55	55	55	55	55
Size of pump		2	2½	2½	3	3	3	3½	3½	4	4	4	4
Size of suction		2½	3	3	4	4	4	5	5	6	6	6	6
H. P. of motor		5	7½	7½	7½	10	10	15	15	15	15	15	15
Fresh water connection		¾	¾	¾	¾	¾	¾	1	1	1	1	1	1
Drain connection		2	2	2	2	2	2	2	2½	2½	2½	2½	2½
Area, sq. ft.		56	70	84	98	112	126	140	154	170	184	198	212
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all dimensions	A	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″	11′ 2″
	B	6′ 1″	7′ 7″	9′ 1″	10′ 7″	12′ 1″	13′ 7″	15′ 1″	16′ 7″	18′ 1″	19′ 2″	21′ 1″	22′ 7″
	C	11″	12″	12″	14″	14″	14″	16″	16″	16″	16″	16″	16″
	D	46″	47½″	47½″	47½″	51¾″	51¾″	51¾″	55¾″	55¾″	59¾″	59¾″	59¾″
	E	37″	37″	39″	41″	41″	41″	41″	41″	42″	42″	42″	42″
	F	5′10″	7′ 4″	8′10″	10′ 4″	11′10″	13′ 4″	14′10″	16′ 4″	17′10″	19′ 4″	20′10″	22′ 4″
	G	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″	9′ 6″
	H	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″

CAPACITY OF SIZE "F," TYPE "A," AIR PURIFIERS

Number of Purifier		F-4	F-5	F-6	F-7	F-8	F-9	F-10	F-11	F-12	F-13	F-14	F-15	F-16	F-17
C. F. M.		51,000	59,500	68,000	76,500	85,500	94,000	102,500	111,500	120,000	128,500	137,000	145,500	154,000	161,500
Gallons per minute		240	288	336	384	432	480	528	576	624	672	720	768	816	864
Total head		57	57	57	57	57	57	57	57	57	57	57	57	57	57
Size of pump		2½	3	3	3½	3½	4	4	5	5	5	5	5	5	5
Size of suction		3	4	4	5	5	6	6	7	7	7	7	7	6	6
H. P. of motor		7½	10	10	15	15	15	15	15	20	20	20	20	20	25
Fresh water connect'n		¾	¾	¾	¾	¾	¾	1	1	1	1	1	1	1	1
Drain connection		2	2	2	2	2	2	3	3	3	3	3	3	3	3
Area, sq. ft.		102	119	136	153	171	188	205	223	240	257	274	291	309	332
Length over all		6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″	6′ 9″
Over-all Dimensions	A	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″	13′ 2″
	B	9′ 1″	10′ 7″	12′ 1″	13′ 7″	15′ 1″	16′ 7″	18′ 1″	19′ 7″	21′ 1″	22′ 7″	24′ 1″	25′ 7″	27′ 1″	28′ 7″
	C	12″	14″	14″	16″	16″	17″	17″	17″	18″	18″	18″	18″	18″	18″
	D	47½″	51¾″	51¾″	57¾″	57¾″	59¾″	59¾″	67″	67″	67″	67″	67″	67″	67″
	E	39″	41″	41″	41″	41″	46″	46″	48″	48″	48″	48″	48″	48″	48″
	F	8′10″	11′ 4″	12′10″	14′ 4″	15′10″	17′ 4″	18′10″	20′ 4″	21′10″	23′ 4″	24′10″	26′ 4″	27′10″	29′ 4″
	G	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″	11′ 6″
	H	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″	14″

Materials Used in Air Washer Construction. The materials ordinarily used in air washer construction in the order of their desirability, are:

Cold rolled copper.

Galvanized American ingot iron or toncan metal.

Galvanized steel.

For filling in the blank spaces in specification forms, the following recommendations are made for copper and galvanized air washers:

Air Washer	Cold-Rolled Copper	Galvanized American Ingot Iron Galvanized Steel
Casing	18 oz. up to 15,000 cu. ft. 20 oz. on larger sizes	No. 18 U. S. Std. Gage
Eliminator	Same as above	No. 20 U. S. Std. Gage
Tank	40 ounce Copper	No. 12 U. S. Std. Gage
Angle Braces	Copper-Plated Angle Iron 1 1/2″ x 1 1/2″ x 3/16″ to 45,000 C. F. M. 2″ x 2″ x 3/16″ on larger sizes	Galvanized Angle Iron Sizes: See opposite column
Flanges	Brass	Galvanized Malleable Iron
All piping	Brass	Galvanized Iron

Other gages and kinds of materials may be specified, if desired, but those embodied in the above table are based on a wide experience.

Concrete Tanks. Frequently on account of low headroom or for other reasons, it becomes more desirable to use a concrete tank instead of sheet metal.

Deflectors. It is desirable that air entering should be fairly equally distributed over the spray chamber area and, whenever required, provision should be made for the necessary deflectors to equalize the distribution.

TYPICAL AIR WASHER AND HUMIDIFIER SPECIFICATION

Air Washer. Furnish and erect where shown on plans, one (1) .... "Type........" air washer of...........cubic feet per minute capacity, including spray chamber, mist nozzles, eliminator, water tank, centrifugal pump with direct-connected electric motor; the entire apparatus to be erected complete in accordance with plans and instructions.

The over-all dimensions of the Air Washer are to be not less than the following: height, ...; width, ...; length, ...; velocity of air through gross area of spray chamber not to exceed 500 feet per minute.

Casing. The casing to be constructed of......................................., firmly braced with ....angles, spaced not over 3′–0″ apart. Provision must be made to prevent "back-lash" of mist from spray chamber. The necessary................ flanges for pipe connections to casing shall be riveted and soldered to the same.

A 16″ x 30″ glass-paneled, cast-iron door and frame is to be placed in the casing for access to, and inspection of, the interior; the door to be held shut by at least two cams and made watertight by the use of suitable packing.

Spray Chamber. The spray chamber is to contain a series of mist nozzles for atomizing the spray water. The nozzles in the main spray system are to be arranged uniformly over the entire spray chamber area at a distance of not less than 4′–3″ from the eliminator baffles, and are to discharge in the direction of air flow. The nozzles are to be tapped into risers of ample size, said risers to be tapped into a main header.

To insure a uniform density of mist within spray chamber, and also the proper atomization of spray water, each nozzle in the main spray system is to handle not more than 1¼ gallons of water per minute and to operate at a pressure of not less than 20 pounds per square inch.

In addition to the main spray system there is to be provided a series of mist nozzles, to be operated at 5 pound pressure for flooding the eliminator baffles in humid summer weather, so that fair air cleansing results may be obtained by their use alone, without any appreciable addition of moisture to the passing air. The contractor is to place in the supply connections to the main spray system and flooding nozzles 3½″ nickel-plated water-pressure gages which are to be furnished with the air washer.

The total water volume discharged from all mist nozzles is to be not less than........... gallons per minute under the specified operating conditions.

All mist nozzle piping and headers are to be constructed of..........................pipe; mist nozzles are to be all-brass, and each is to have a discharge orifice not less than 3/16″ in diameter.

Eliminator. For eliminating free moisture from the air after it has been subjected to the sprays, a series of vertical baffles are to be provided; baffles to be constructed of.............. ....rigidly braced as required.

In order that the entire eliminator surface may be accessible for cleansing and inspection, the baffles are to be spaced 3″ on centers, and shall deflect the air not more than four times, each deflection to be at least 5″ in length and no single angle of deflection to exceed 30°; the front and rear sections of the eliminator to be staggered so that the air streams will be split in their passage through the same.

Tank. The tank is to extend under the entire air washer and is to be constructed of........ ....firmly braced with................angles; all joints to be riveted and soldered.

The necessary...........................flanges are to be riveted and soldered to the tank for all necessary pipe connections to the same.

Within the tank is to be located a bell-mouth overflow fitting, automatic fresh water supply valve and a completely submerged strainer built of perforated brass, covering the outlet to the pump suction.

Strainer to be of the double box type with hinged lid and a removable inner basket having a free area at least twenty (20) times that of the pump suction. The entire strainer is to be easily removable for cleansing and inspection.

With the air washer is to be furnished a suitable length of hose with fittings for attachment to ball float valve, for cleansing and flushing out the apparatus.

Pipe Connections. The following, and all other outside pipe connections to Air Washer, with necessary valves and fittings, are to be furnished and put in place by the contractor in accordance with plans and instructions; long sweep fittings to be used in the connections between pump and air washer.

.... Pump discharged connection to main sprays.
.... Supply connection to flooding nozzles.
.... Connection between pump and strainer in tank.
.... Fresh water supply to ball float valve.
.... Overflow and drain from tank to sewer.
.... Priming connection to pump.

The two 3½″ nickel-plated water-pressure gages furnished with the air washer are to be located by the contractor as directed; all connections to be made of................pipe.

Pump. A centrifugal pump of ample capacity to maintain the proper pressure on the nozzles is to be provided for recirculating the spray water between the tank and the mist nozzles.

The pump is to be provided with split casing, so that the interior may be removed without disturbing pipe connections

The pump is to be direct-connected by means of a flexible coupling to a.......(Make)......
............h.p., ...volt, ...current, ...phase, ...
cycle electric motor, both to be mounted on a common cast-iron base provided with oil rim around the same; motor to be equipped with suitable starting device.

The pump and motor are to be placed on a firm foundation; bed plate to be wedged up at the bolt holes and leveled before bolting down, and when in perfect alignment, to be properly grouted in.

Guarantees. The manufacturer is to guarantee the apparatus furnished to be first-class in every respect, built of the best material and workmanship, and is to agree to replace any part showing defects due to defective material or workmanship which may become apparent within a period of one year from date of shipment.

It is to be further guaranteed that the apparatus, if erected and operated in accordance with plans and instructions, will remove entirely all entrained water and free moisture from the air passing through the air washer, and will remove ninety-eight per cent (98%) of all solid matter contained in the entering air.

The apparatus is to be guaranteed during the summer to reduce the temperature of the entering air not less than eighty-five per cent. (85%) of the initial wet-bulb depression when recirculating the spray water.

The air washer is to be guaranteed to offer a resistance of not more than 0.25 of an inch water gage to the air passing when the air washer is operating at the rated capacity.

DRYING MATERIALS

The *artificial drying* of various materials in order to accelerate the process of manufacture has, in late years, become an important branch of mechanical engineering. Materials are dried by passing warm and comparatively dry air over the exposed surfaces, the air becoming more or less saturated by evaporation during its passage through the drier. The *process is adiabatic*, the heat required for evaporation being supplied by the lowering of the air temperature similarly to the process previously described under "Air Cooling by Evaporation."

It is observed in practice that the *wet-bulb temperature is constant* in practically all parts of the drier and equal to the wet-bulb temperature of the entering air.

The temperature at which *evaporation takes place is the wet-bulb temperature.*

The lowest temperature possible for the air leaving the drier will evidently be the wet-bulb temperature of the entering air as the air becomes adiabatically saturated at this temperature. In practice the air never leaves in a saturated state. It is usual in drying problems to assume the percentage of relative humidity of the leaving air, which experience with the class of material being dried and upon its distribution and exposure to the air has demonstrated, may be obtained. Having assumed the final percentage of humidity the corresponding temperature of the leaving air is readily determined by means of the psychrometric chart (Fig. 2).

Weight of Air to be Circulated per Hour.

Let N = weight of water to be evaporated per hour.
t = outside air temperature.
t_1 = temperature of heater air entering drier.
t_2 = temperature of the leaving air. This temperature is fixed by the per cent relative humidity assumed for the leaving air.
W_1, W_2 = weight of vapor carried by one pound of entering air and leaving air respectively.
B = weight of dry air to be circulated per hour.

Then $$\frac{N}{W_2 - W_1} = \text{weight of air to be circulated per hour, lb.}$$

Example. If the outside air temperature t is 80° F. and 40 per cent relative humidity ($63\frac{1}{2}$° F. wet bulb) and the temperature of the air t_1 introduced into the drier is 110° F., and assuming that the material to be dried is so exposed that the air will leave the drier 70 per cent saturated, required the leaving air temperature t_2 (Fig. 6).

Referring to the psychrometric chart—from the intersection of the vertical 80° temperature line and the 40 per cent humidity curve, follow the diagonal *constant weight of vapor line* until it intersects with the vertical 110° line which will correspond to a relative humidity of approximately 15 per cent, read horizontally back to the left to the intersection of the 70 per cent humidity curve, the corresponding temperature being 81° F., and is the temperature required (wet bulb temperature 73° F.). The weight of moisture evaporated per pound of dry air circulated is the difference between the weight of vapor per pound of dry air for 81° F. and 70 per cent relative humidity and 80° F. and 40 per cent relative humidity or 0.0159 − 0.0088 = 0.0071 lb.

Example. Let it be required to determine the weight of air to be circulated through a drier for the following assumed conditions of operation: Weight of water to be evaporated per hour N = 6000 lb. The temperature t and relative humidity of outside air 60° F. and 60 per cent, respectively. Temperature of air introduced into drier, $t_1 = 200°$. $W_1 = 0.00654$ weight of vapor per pound of air. As the initial temperature t_1 is beyond the range of the chart it is necessary to make a calculation to determine the temperature t_2

The amount of heat required to raise one pound of dry air from 0° F. to 200° F. and evaporate the weight of moisture contained by the air, or 0.00654 lb., is found as follows:

This weight corresponds to a dew-point temperature of 45° F., the total heat required is therefore $0.00654 \times 1{,}065$ (latent heat at 45° F.) $+ 0.24 \times 200 + 0.44 \times 0.00654\ (200-45)$ or 55.4 B.t.u.

The last term is the heat required to superheat the vapor from the dew point to the final temperature and may be neglected without much error.

Locating 55.4 B.t.u. on the right hand side of the chart it is seen that the air would become adiabatically saturated, by reading horizontally across to the saturation curve, at 91½° F. which will be the wet-bulb temperature in the drier and for 70 per cent saturation the dry-bulb temperature is 100.5° F., which is the temperature t_2 of the leaving air. The weight of vapor carried per lb. of air is therefore 0.043×0.70 or $W_2 = 0.0301$ lb. The evaporation takes place at the wet-bulb temperature.

The weight of air to be circulated per hour is,

$$\frac{N}{W_2 - W_1} = \frac{6000}{0.0301 - 0.0065} = 254{,}237 \text{ lb.}$$

The size of the heater required is readily determined from the data given in the chapter on "Hot Blast Heating." The duty of the heater being to raise the temperature of the incoming air from 60° F. to 200° F. plus about 10° to allow for the loss in the underground ducts. The weight of steam necessary at 5 lb. gage pressure per hour is:

$$W = \frac{254{,}237 \times (200 - 60) \times 0.24}{960} = 8{,}900 \text{ lb. per hour}$$

corresponding to about 300 boiler horsepower.

To make up for the radiation loss from the drier and heat up the material introduced a higher initial temperature t_1 will be necessary, requiring a somewhat greater boiler capacity.

The Design of Drying Equipment. The following is an extract from a paper on "Drying Apparatus," *Trans. A. S. H. & V. E.*, 1912, by *H. C. Russell:*

"The manner of constructing the drying tunnels or compartments, the amount of air to be circulated, the temperature of this air, and indeed almost every detail of design, depend upon the material to be dried and its amount. For different classes of material these details will vary over a wide range. Some materials are very sensitive as regards air conditions in the drier and others will stand considerable abuse. In progressive drying the green wet material should be introduced into the drier at the opposite end from the hot air entrance. By this method the material is dried gradually and is not so liable to case harden on the outside and retain a portion of its moisture."

Temperatures. It has been found that for practically each material the maximum temperature in the drier for good results should not exceed a given figure and for temperatures much below this figure in the hottest part of the drier the capacity will either be reduced or the material will not dry at all. Some of these temperatures are given in tabular form hereafter.

Drying Period. For each material there is a minimum drying time which, of course, varies very widely for different materials. If we try to extract the moisture from any given material

in much less than this time we will get unsatisfactory results due to too rapid evaporation. Some drying periods are given in tabular form hereafter.

Humidity. It will generally be found when hot air is first brought into contact with the material which is almost dry (the air gradually absorbing moisture as it passes along toward the wet end of the drier and the fresh material) that no trouble will result from too rapid evaporation unless the air supply is too great or the temperature too high.

Generally, if calculations are based on about 60 per cent relative humidity at 50° F. outside temperature the result will be on the safe side, because when the outside temperature is below 50° F. one can dry with a smaller volume of air due to the fact that the moisture absorbing capacity of the air will be greater, and when the temperature rises above 50° F., the relative humidity will probably be lower than 60 per cent, or the temperature of the drier may be increased somewhat. The relative humidity and temperature can be calculated from the above assumed conditions. At the green end the relative humidity differs greatly for different classes of work. Very seldom can it be made to exceed 75 per cent at the temperature of the air leaving the drier.

Moisture. The amount of moisture to be evaporated in a unit of time must always be known for intelligent design. This can be found by simple experiment if in no other way.

Air Supply. The air supplied in dropping from the temperature of the incoming to that of the outgoing air must give off sufficient heat to take care of the following heat losses: (a) Radiation from the drier. (b) Heat required to raise the temperature of the products to be dried, the water to be evaporated and the trucks, etc., from factory temperature to drier temperature. (c) Heat required to evaporate the moisture based upon the latent heat of evaporation at the temperature at the wet end of the drier. The air may be safely admitted to the drier at a temperature above that given in the table hereafter to offset all losses except the last named.

There are *two requirements* as to air volume which must be met: (a) There must be sufficient volume of air as a vehicle for carrying sufficient heat to provide for all the above heat losses without reducing the temperature at the green end too low for good results. (b) There must also be a sufficient weight of air to carry away the moisture without coming too near the point of saturation. This latter is to be calculated on a basis of the difference in the amount of moisture contained in a cubic foot of air as delivered to the drier and that contained in the same air as it leaves the drier, the relative humidity at the green end never to exceed 75 per cent.

Loss of Temperature. After allowing for all heat losses except those required to evaporate the moisture, the further loss of temperature in the drier bears a definite ratio to the amount of moisture we can require each pound of air to carry away, which, of course, depends upon the relative humidity at the discharge end. In the ordinary waste heat brick drier this loss is about 100 degrees; in lumber driers, it varies from 30 to 70 degrees, and for the majority of other classes of driers it runs from 10 to 30 degrees.

The accompanying table gives the maximum temperatures, and in some cases the average drying period, for a few materials. Driers for lumber and brick have by reason of their commercial importance been given considerable attention and very reliable data have been secured:

TABLE 9

CONDITIONS FOR DRYING DIFFERENT MATERIALS

MATERIAL	Temp. Deg. F.	Drying Period
Sole leather hides	90	4 to 6 days
Thin leather hides	90	2 to 3 days
Bone glue	70 to 90	4 days
Skin glue	70 to 90	2 days
Starch	180 to 200	12 hr.
Apples	140 to 180	6 hr.
Leaf tobacco	85	......
Stem tobacco	200	......
Soap	100	2 days
Wood	105	......
Rags	180	......
Pottery	120	......

In many drying installations waste heat can be utilized for the purpose. All brick and tile yards of modern design are now drying their products from waste heat derived from kilns cooling down after being burned, and it so happens that if the rotation of kilns is properly studied and adhered to there is waste heat available practically at all times in about sufficient amount for drying the fresh product with a liberal allowance for waste. Much more can be accomplished in the way of adapting driers to use waste heat, and furthermore the air can generally be drawn directly over the cooling surfaces themselves, for we are seldom interested in the absolute purity of the air supply to such driers.

CHAPTER XVII

TEMPERATURE AND HUMIDITY CONTROL

AUTOMATIC REGULATING APPARATUS

The regulation of temperature and humidity in buildings equipped with heating and ventilating systems is most satisfactorily accomplished by the use of automatic thermostats and humidostats, which after having been set to maintain the desired degree of heat or relative hu-

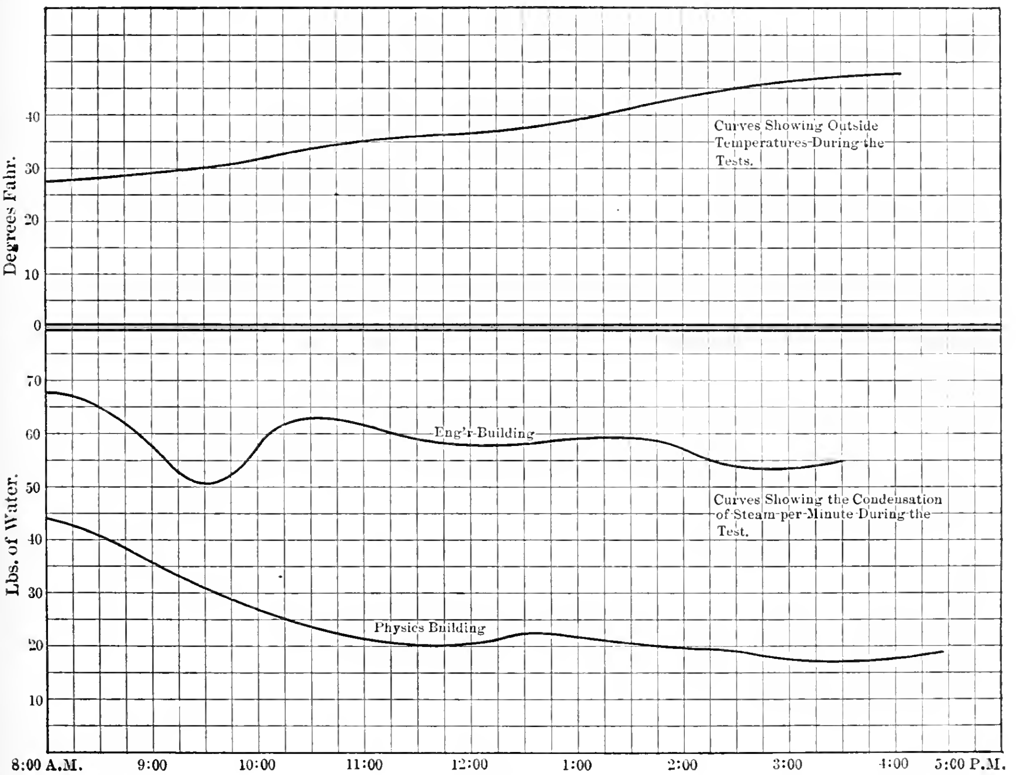


FIG. 1.

midity, automatically control the valves or dampers throughout the heating system in such a manner as to keep these conditions practically constant.

Economy of Temperature Control Systems. The desirability and economy of automatically maintaining constant and reasonable temperatures and humidities have been repeatedly proven by comparing records of attendance, work performed, steam or coal consumption, etc., in the same buildings before and after such an installation, or in different buildings of the same

size and construction. The chart (Fig. 1) shows an example of this sort where the steam consumption of two buildings at the University of Illinois of practically equal size has been measured and plotted for a typical day. The uniform reduction in the steam consumption of the thermostatically controlled Physics building as the outside temperature rose is most significant. The economy of temperature control is also readily shown in a case where overheating would result without it. A certain building is kept at 75° instead of 70° F., with an average outside temperature of 40° F. The heat supplied, compared to that required, is in the ratio of 35/30 = 1.17, or 17 per cent is wasted.

Such a saving in fuel may be confidently expected in a building thermostatically equipped as compared with the usual manual control of the heating equipment in the same building.

Small Plants. Temperature regulating systems may operate to control the entire heating system from *one central point*, as in the case of small plants, where the regulation of the drafts at the boiler or furnace will give a fairly uniform average temperature for the entire building. In this case the location of the control point is of prime importance, and it must be so selected

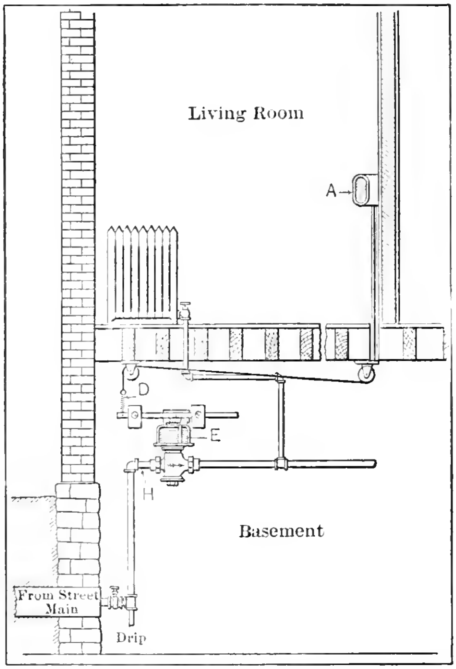


Fig. 2. Regitherm and Reducing Valve on City Steam Main.
A, Regitherm; *D*, Spring; *E*, Reducing Valve; *H*, Steam Pipe from Street Main.

as to represent the average temperature conditions as nearly as possible. The simplest system of this type (Fig. 2) operates directly on the steam supply to the building, opening or closing the reducing valve in accordance with the temperature existing at the thermostat *A* in the living room. In case the building is equipped with a separate plant the regulator is simply connected to the supply and check drafts, and thereby controls the intensity of the fire so as to maintain a constant temperature at the thermostat. Thermostats are also made which produce very little power in themselves, but, due to their action, are able to release a powerful spring or heavy weight which is capable of opening or closing the necessary dampers to secure regulation of the fire.

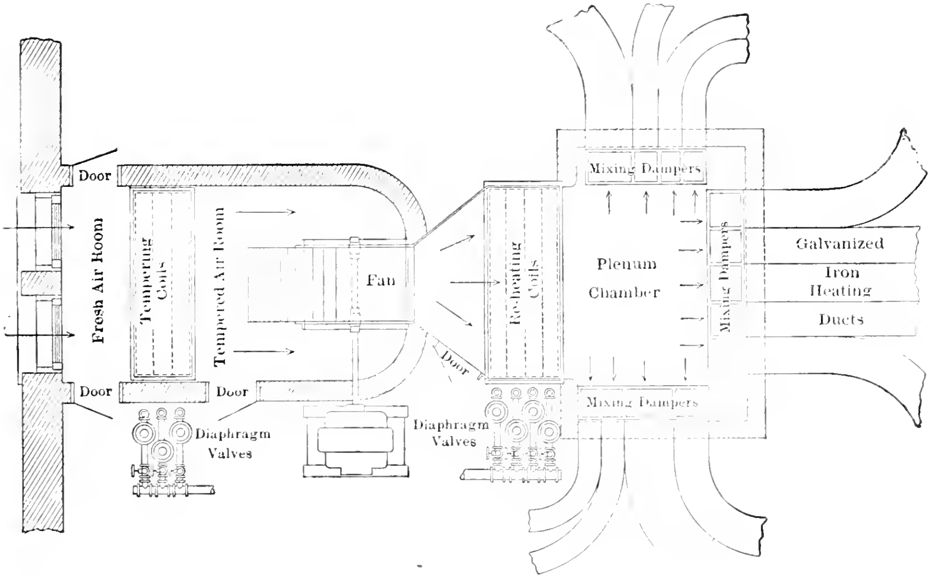


Fig. 3.
Plan View of Indirect System
(See Fig. 4)

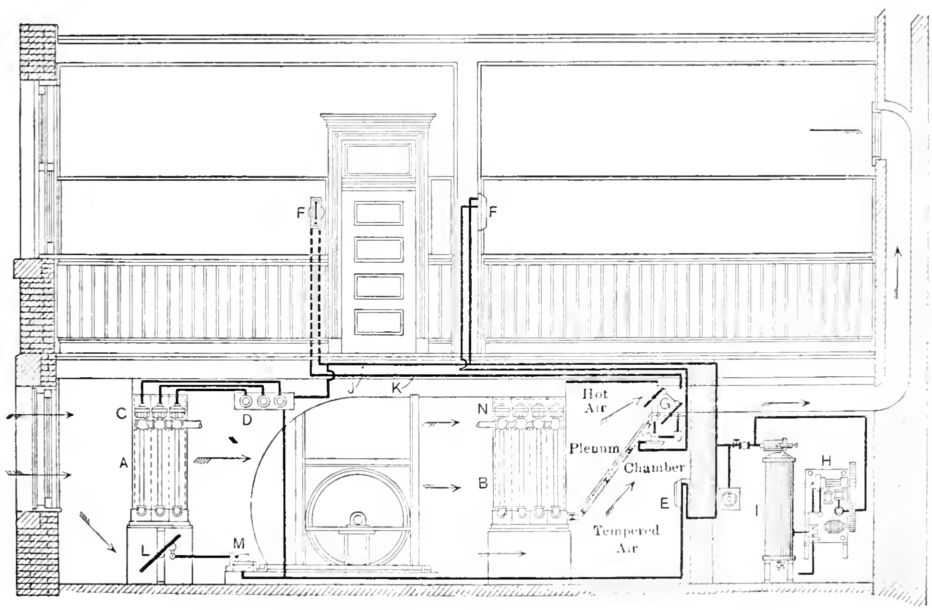


Fig. 4. Powers System of Regulation—Indirect System.

Large Plants. Temperature regulating systems for large installations, where it would be impossible to find one central point representing a fairly uniform average temperature for the entire system, consist of *individual thermostats* located in each room. These thermostats either operate through the medium of compressed air to close or open the control valves on the radiators, or to change the position of the dampers controlling the mixture of warm and tempered air entering the room. One thermostat, in large rooms, should be allowed to control not more than 2 or 3 radiators, although a single compound thermostat may be used to control both the entering air temperature and the direct radiation in the case of combination systems.

Typical Systems for Large Plants. The application of thermostatic control to a large indirect heating system with mixing valves supplying warm and tempered air through a single duct system is shown in Figs. 3 and 4. Thermostats F are located in each room, and supply

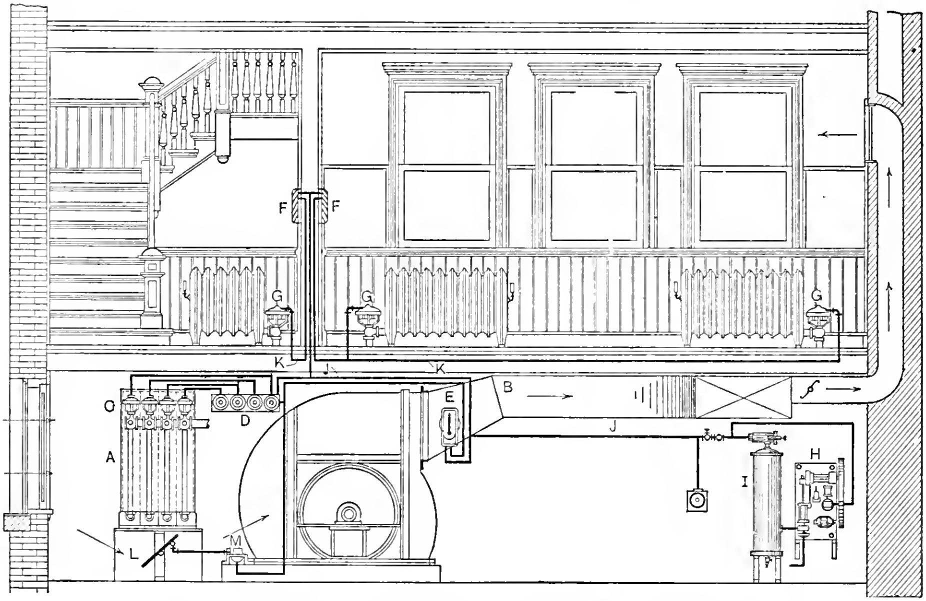


FIG. 5. POWERS SYSTEM OF REGULATION—COMBINATION SYSTEM.

compressed air to the damper motors which are linked to the mixing dampers G controlling the proportions of hot and tempered air entering into the constant volume of air supplied to each room. Since the tempered air must be kept at from 60° to 65° F. a thermostat E is located in the tempered air room, and supplies compressed air at varying pressures to the relays D which close or open the diaphragm steam valves C on the tempering coil A in succession in accordance with the variations in the outside air temperature. This same thermostat also operates the cold air by-pass L under this coil, and in this way gives a more gradual temperature control than could be obtained by cutting out a whole section of coil at once. The reheating coils B may be either manually or thermostatically controlled by valves N.

The compressed-air supply is obtained from the air storage tank I in which a pressure of 15 lb. is automatically maintained by an electrically driven air compressor H. From the storage tank an air supply line J must run to each thermostat and relay valve, and another line K is then run from the thermostat or its relay to the diaphragm valve or damper motor.

Thermostatic control as applied to a combination direct heating system with positive ventilation is shown in Fig. 5. In this case the room thermostats F operate on the supply valves

on the direct radiators and a single duct thermostat controls the indirect coils and by-pass under same. The indirect coils are controlled successively by the relays D, and the compressed air is supplied by compressor H and tank I as for the simple indirect system.

THE THERMOSTAT

Method of Operation. The essential feature of any thermostatic control system is the thermostat itself, which must so change in shape or pressure under very slight changes in temperature—the range being about 1° or 2° above or below the standard desired—that it will, either

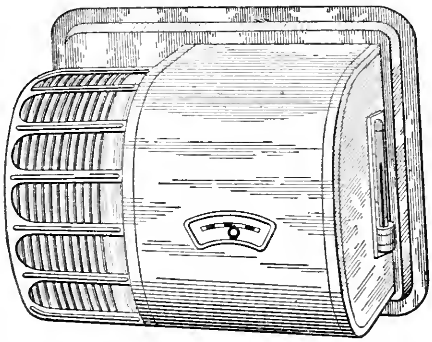


Fig. 6. Regitherm with Sylphon Bellows. (*American Radiator Co.*)

directly, or *indirectly* by use of some other medium, open or close the necessary valves or dampers.

Thermostats are also classified as (1) *positive*, and (2) *intermediate* or gradual movement type. The former move through their entire range instantly and cause a similar motion in the valve or damper from fully open to fully closed, while the latter are so designed as to give a gradual motion to the valve or damper, and then hold it firmly in an intermediate position, so long as such position will produce the desired temperature at the thermostat. The latter type must *never* be used with steam apparatus connected on the one pipe system, or hammering and flooding will result.

Direct Acting Thermostats. The *Regitherm* (Figs. 2 and 6) is of the direct type and, due to the volatilization of the low boiling liquid within the brass bellows, develops a pressure of 15 lb. for a change of 1° F. and gives a movement of ½″, which is multiplied by a suitable lever and transmitted to the valve or damper lever by a small wire cable. This apparatus is of the gradual movement type.

The majority of thermostats operate indirectly, the expanding element in the thermostat operating to admit air or water under pressure to a diaphragm which is of proper size to exert the required pressure at the valve or damper and thus close or open it. If the pressure thus transmitted operates to close the valve, then either springs or weights of the proper tension or amount are used to open the valve, whenever the air or water pressure is released by the thermostat.

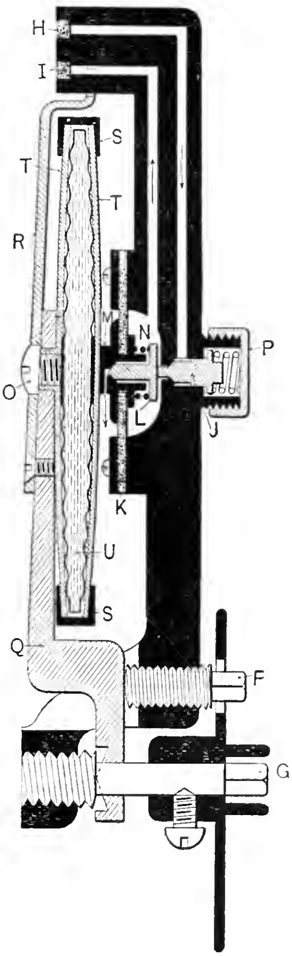


Fig. 7. Powers Thermostat "K"

Indirect Acting Thermostats. The three most common commercial systems are the *Powers*,

Johnson, and *National*, each of which uses a different expansion element in the thermostat itself, but they all use the same general system of control, and the same specialties and accessories. All of these systems use compressed air at about 15 lb. pressure as the operating medium.

The *Powers Thermostat* (Fig. 7) is a reducing valve in principle, and is of the gradual movement type, the essential feature being a flexible thermostatic disc T filled with a low boiling liquid. Under the influence of a rise in temperature of the surrounding air the disc expands due to volatilization of the liquid U. This motion flexes the diaphragm K and opens valve J, admitting air from the inlet H into chamber N, and thence through outlet I to the diaphragm valve at the radiator, closing same; or if full pressure is not admitted, partially closing the valve. The escape valve L is held tightly against its seat at this time. If the room temperature drops the thermostatic disc contracts and the compression spring above valve J promptly closes same, cutting off the high pressure air from the supply line H. Diaphragm K returns to its normal position and the compression spring under valve L lifts it from its seat, opening up the escape port in M so that the pressure in chamber N drops to atmospheric, and springs or weights open the diaphragm valve at the radiator or the damper motor.

Adjustments in the operating range are made by use of screws F and G, which change the position of the thermostatic disc frame with respect to the valve housing. The thermostatic

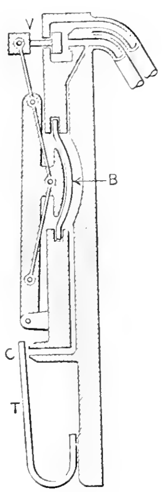


Fig. 8.
Radiator Valve Closed.

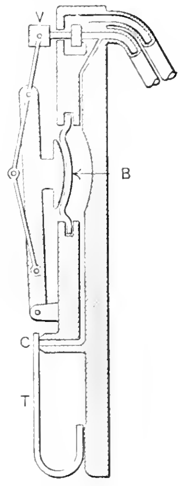


Fig. 9.
Radiator Valve Open.

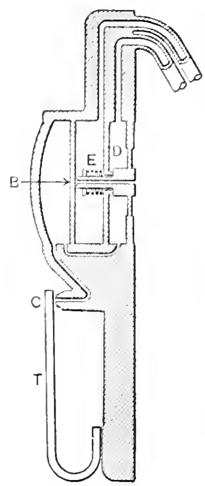


Fig. 10.
Holding Hot Air Damper in Fully Open Position.

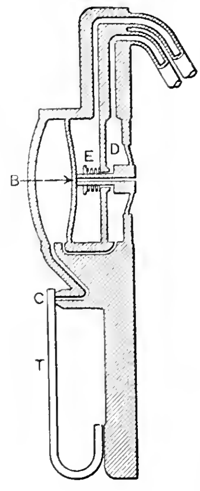


Fig. 11.
Operating to Close Hot Air Damper.

Johnson Positive Thermostat.*

Johnson Intermediate Thermostat.*

element operates directly without levers on a double valve controlling both supply and waste and uses only the air necessary to inflate the diaphragms.

The *Johnson Thermostat* (Figs. 8 to 11) is made up in two forms, one for positive service, (Figs. 8 and 9) throwing the full air pressure instantly into operation, and the other for intermediate service, (Figs. 10 and 11) acting more or less like a pressure reducing valve, and holding valves or dampers in intermediate positions. The thermostatic element is a compound metallic strip of brass and steel, which in expanding and contracting opens and closes a small port, thereby creating or releasing pressure for operating the air valves in the thermostat.

In the accompanying cuts, the curved metal strip T is the element affected by the room

* Note.—"*The Johnson Service Co.* has recently brought out a new type of positive thermostat in which the knuckle movement shown above is much simplified, and the size of the instrument materially reduced. The principle of operation is essentially as here shown.

temperature. A slight change in temperature immediately affects the strip and the movement causes it to either open or close the small air port C. When this port C is open, a small amount of air escapes, but when the strip closes the port, it causes a pressure to collect on the diaphragm B. In the *positive thermostat* this pressure forces out the knuckle movement, which, when it passes the center position, instantly pushes in the valve V. This movement of valve V immediately releases the air pressure in the branch line, permitting the radiator valve to open. When

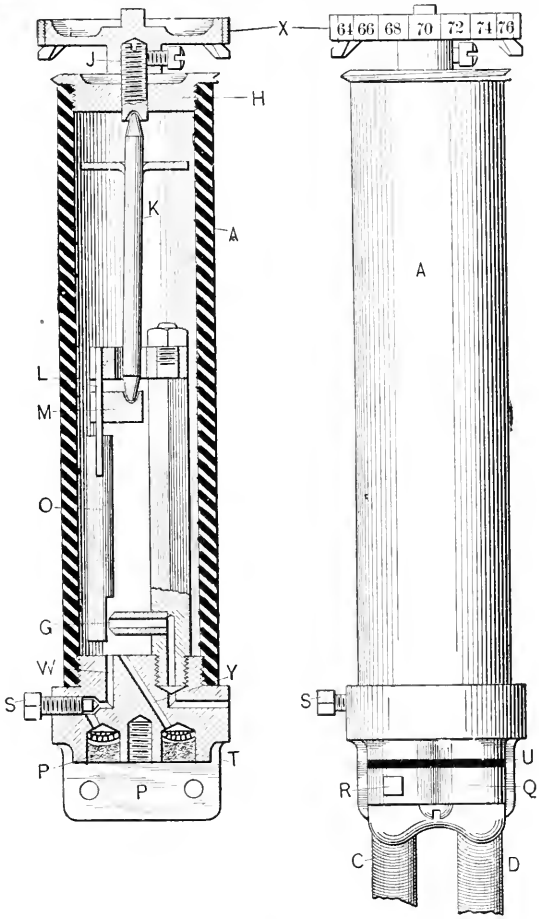


FIG. 12.
NATIONAL POSITIVE TYPE OF THERMOSTAT.

T expands outward, the air pressure on B is relieved, the valve V is instantly thrown outward, and the full air pressure is at once turned into the branch radiator valve. The diagrams show the thermostat in each position.

In the *intermediate thermostat* the thermostatic strip T moving inward or outward, as affected by the room temperature, varies the amount of air which can escape through the small port C. When the port C is completely closed the full air pressure collects on the diaphragm B, which forces down the main valve, letting the compressed air from the main pass through the chamber D into chamber E, as the valve is forced off its seat. The air from chamber E passes into the branch to operate the damper.

When port C is fully open, the air pressure on diaphragm B is relieved, the back pressure in E lifts up the diaphragm, and the air from the branch escapes out through the hollow stem of the main valve. The thermostat thus operates on a reducing valve principle which insures various pressures as required in the branch line to operate the damper.

Air from the high pressure supply mains enters the *Johnson* thermostat through the lower tube at the top and passes to the diaphragm valve or motor through the upper tube, and both tubes are run concealed in the wall or floor construction.

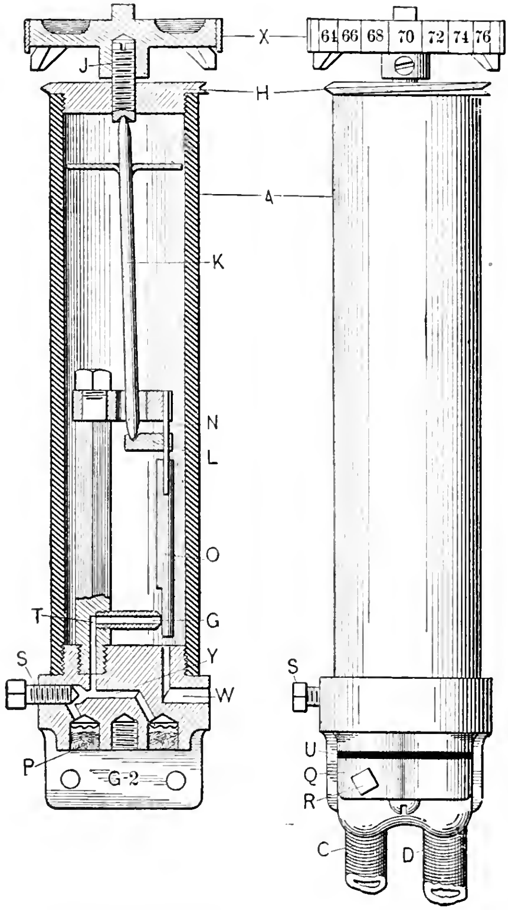


Fig. 13. The National Intermediate or Graduated Type of Thermostat.

The *National Thermostat* is of the expansion type, and is made for both positive (Fig 12) and graduated (Fig 13) service.

The *positive type* of thermostat consists of the thermostatic tube A of vulcanized rubber, a change in the length of which produces motion in the valve lever O by means of the plug H and the connecting rod K pressing on the seat M. This valve is permitted to open the port G under increased pressure on M by the flexure of the plate spring L and, when this pressure is relieved, the spring L causes the valve to close. Air under pressure is supplied to the thermostat by the

pipe *C*, connected to the air supply passing into the thermostat through the filter *P* and the restricted passage *W*, and thence to the valve through the port *Y*, the filter *T* and the pipe *D*. The adjustment of the thermostat for different temperatures is provided for by the screw *J* through the top plug, and the indicating disc *X*. The screw *R* in the connector *Q* at the base of the thermostat is a needle valve which opens or closes connection with the air supply, and is used as an air shut-off valve in case it is desired to remove the thermostat. The screw *S* is a similar valve which controls the supply of air to the thermostat, and this screw is set so as to allow the air to pass into the thermostat in a restricted quantity.

When the temperature of the apartment has risen so as to expand the thermostatic element *A*, the pressure on *K* and *M* is relieved, and the spring *L* closes the port *G*. The air admitted through the passage *W* passes through the port *Y* and the pipe *D* to the diaphragm of the valve, and closes the heat supply to the radiator. When the temperature of the apartment has fallen so as to produce contraction of the thermostatic element *A* the pressure on *M* of the connecting rod *K* moves the valve lever *O*, thus opening port *G*. This allows the air which has accumulated above the diaphragm of the valve to return through the port *Y* and escape through the port *G* to the atmosphere, permitting the valve to open. The restricting screw *S* must be so adjusted that when *G* is open the diaphragm valve will open positively and promptly.

The *graduated type* of thermostat consists of the thermostatic tube *A* of vulcanized rubber, a change in the length of which produces motion in the valve lever *O*, by means of the plug *H* and the connecting rod *K* pressing on the block *L*. This valve lever is permitted to open the port *G* under increased pressure on *K* by the flexure of the plate spring *N*, and when this pressure is relieved this spring *N* causes the valve to close.

Air under pressure is supplied by the pipe *C* connected to the air supply, flowing into the thermostat through the filter *P*, the restriction *S*, the passage *T* and the port *G*. The adjustment of the thermostat for different temperatures is provided for by the screw *J* through the top plug *H* and the indicating disc *X*. The screw *R* in the connector *Q*, at the base of the thermostat, is a needle valve which opens or closes the connection with the air supply, and is used as an air shut-off valve when it is desired to remove the thermostat. The screw *S* is a restriction valve which controls the supply of air to the thermostat, and this screw is set so as to allow the air to pass in a restricted quantity.

When the temperature of the apartment has risen so as to expand the thermostatic element *A*, the pressure on *K* and *L* is relieved and the spring *N* closes the port *G*. The air admitted through the restriction screw *S*, since it cannot escape through the port *G*, accumulates in the passage *Y* and pipe *D*, filling the diaphragm chamber at the valve and moving the damper or valve into the position to decrease the supply of heat. When the temperature of the apartment has decreased so as to produce pressure on the connecting rod *K*, through the contraction of the thermostatic element *A*, the port *G* will be opened by the valve lever *O*, allowing the air in the pipe *D*, together with that which flows through the restricting screw *S*, to escape through the passage *W* to the atmosphere. Thus no air can accumulate in the pipe *D*, and hence the diaphragm valve is permitted to open somewhat. The amount of air released through the port *G* by the valve lever *O* varies the pressure accumulated in the pipe *D* and produces the graduated or intermediate action desired.

Compound Thermostats. The combination of a positive and graduated acting thermostat in a single instrument has been accomplished in the *Powers* and the *Johnson* types of instruments. With this instrument a single thermostat in a room is able to control the direct radiators with a positive action, and the indirect system with a graduated action, so that mixing dampers may be held in intermediate positions as required.

Multiple Thermostats. Multiple control from a single thermostat may be readily secured by the use of suitable *relays* connected to the air line from a single gradual movement type of thermostat. This master unit with its thermostatic element admits air of gradually increasing pressure into the relay main. These relay valves have been set to operate at successive pressures such as 6, 8, 10, and 12 lb., and each relay is connected to a separate diaphragm valve

to which it can admit high pressure air when the relay is subjected to the proper pressure by the controlling thermostat. Such a system is shown in Fig. 4 operating on the tempering coil, each section of which can be cut out successively as the temperature at the thermostat E increases, through a small range of 1 or 2° intervals.

Extended Tube Thermostats. An extended tube type of thermostat (Fig. 14) is required for controlling the air temperatures in ducts, or water temperatures in mains and the expansion element is enclosed in a tube of different metal extending into the air or water current, but with the principal working parts, necessary adjustments, and pipe connections brought to the outside of the duct or pipe.

Covers. Covers for thermostats are usually made of pressed metal (Fig. 15) and are designed and finished to match the architectural requirements of the building. These covers usually carry a mercury thermometer and should indicate whether the valve is "open" or "closed." They may be cast from iron or bronze if more elaborate designs are desired than are usually supplied in pressed metal. The cover should be designed so that reasonable

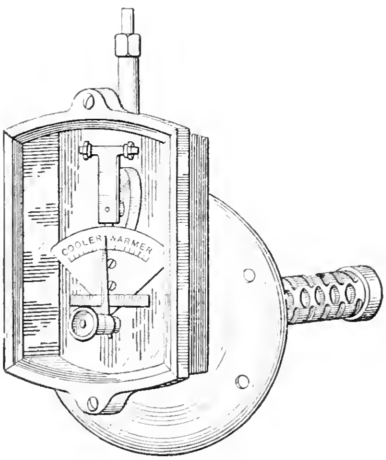


FIG. 14. JOHNSON EXTENSION THERMOSTAT.

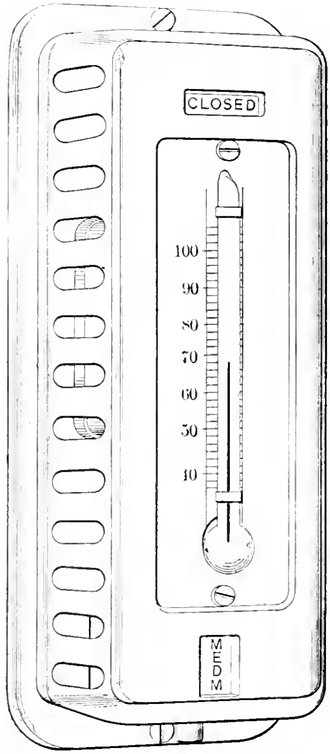


FIG. 15. JOHNSON THERMOSTATIC CASE, 4¾" high, 2" wide and 1" deep.

adjustment of the regulating range may be made without removal of same. A variation of 5° either way, and usually 10° to 15°, is desirable.

The *location* of the thermostat should always be determined by the character of the heating service. Thus, in the case of a directly heated room, occupied by people, the thermostat should be placed on an interior wall 5′-0″ from the floor, and generally not more than two or three radiators should be controlled by one thermostat.

DIAPHRAGM VALVES AND MOTORS

Rubber Diaphragm Valves. Diaphragm valves or diaphragm motors must be connected to all thermostats using air or water as the operating medium. Ordinary angle or globe valves equipped with a special cast iron diaphragm frame (Fig. 16) are usually employed. These valves

must have threadless spindles, and the diaphragms, if they are of *rubber*, must be protected by felt and wood block insulators to which block the spindle is fastened. Air enters chamber *F*, above diaphragm, through inlet *A*, and when sufficient pressure has developed compresses spring *G* closing the valve, which is equipped with a removable composition disc *B*, upon seat *C*. This closes off the steam or water entering *D* and passing to the radiator through outlet *E*. If the pressure in *F* is relieved the spring *G* will open the valve and flow will be re-established.

Iron body valves of larger size for heavy duty are also equipped with pneumatic diaphragms.

Metal Diaphragm Valves. The *Johnson Service Co.* has recently applied the *Sylphon* metal diaphragm or bellows to its thermostatic valves and damper motors to replace the more or less perishable rubber diaphragms. (Figs. 17 and 18.)

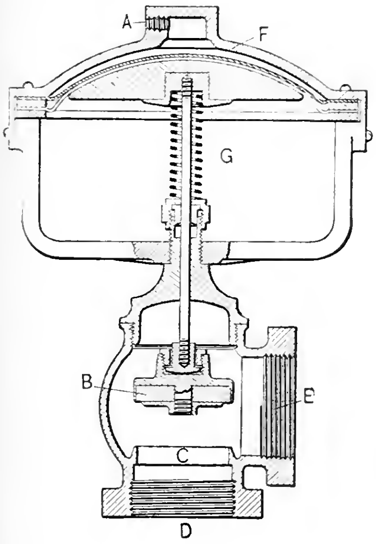


Fig. 16.
Sectional View of Diaphragm Valve.

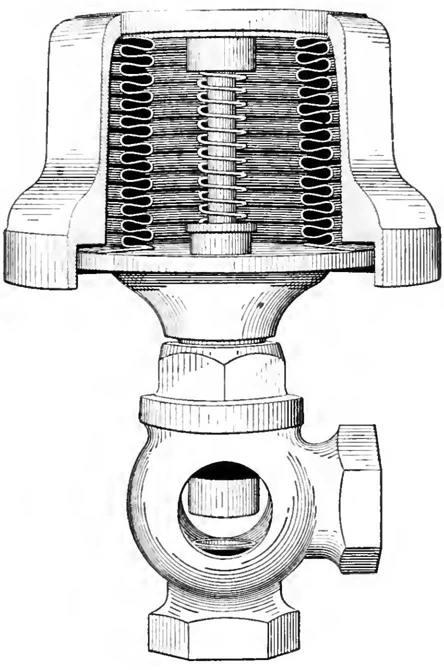


Fig. 17.
Johnson Sylphon Bellows Valve.

This *bellows* is rolled from a single piece of brass with no soldered seams, and the air connection is made at the back of the outside casing, which is air tight. In this way less air is required to operate the valve than would be necessary if the air was supplied to the interior of the bellows. These bellows have been opened and closed as in actual operation over 100,000 times without any apparent effect on the bellows. The Sylphon diaphragms are made in various sizes and the *effective areas* made large enough to operate the valve or damper on 15 lb. air pressure.

The spring around the valve spindle not only serves to open the valve when the air pressure around the bellows is relieved, but also acts to keep the packing around the valve stem constantly tight.

These valves may be supplied with *indicators* to show at a glance when the valve is open or closed, and thus make it possible to check up the performance of the temperature control system at any time.

Diaphragm Motors. These motors (Fig. 19) range from 4″ to 15″ in diameter of diaphragm with double springs above the diaphragm. The large sizes are capable of lifting several hundred pounds. These motors operate the mixing dampers, which are designed to pass a uniform volume of air of any desired temperature. It is very important that these dampers have rigid frames, steel shafts, and stiff close-fitting blades properly braced which must not bind the frame

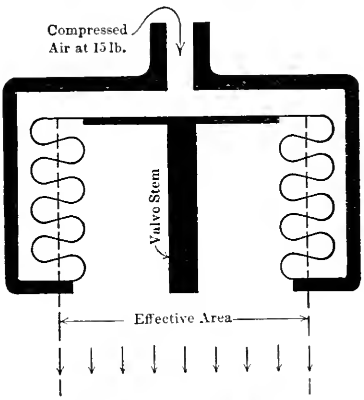


FIG. 18. PRINCIPLE OF JOHNSON SYLPHON VALVE.

at any point. Brass-mounted or ball bearings should be used, and in the larger sizes, where a single blade would be liable to sag or bend out of shape, louvered dampers (Fig. 21) should be required.

Air Compressors. Air compressors, under automatic control, with small storage tanks must be installed in all cases where compressed air at 15 or 20 lb. pressure is not available. This class of service is rather special and the compressors must be adapted to belt, direct acting steam, water or electric drive. Hydraulic and electric outfits (Fig. 22) are by far the most common,

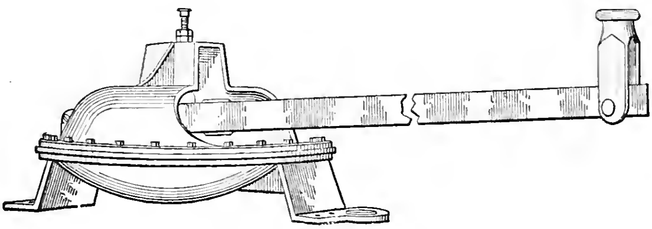


FIG. 19. DIAPHRAGM MOTOR.

and are also the most satisfactory. The size of the compressor and storage tank depends on the number of diaphragm valves or motors installed, and the type of control used at the thermostat, which affects the air consumption.

Remote Control Systems. Compressed air control of distant dampers, by-passes, valves, etc., is being practiced more and more in all buildings where a supply of compressed air is available. For this service it is only necessary to install an air switchboard at some central point, equipped with the required number of pneumatic switches or valves, through which high pressure air may be admitted to the diaphragm motors operating the distant dampers. The position of the damper can be indicated on a graduated dial, and the damper held in intermediate positions if desired.

A simple remote control system is shown in Fig. 23, where switch No. 1 makes it possible to close the fresh air and vent dampers and at the same time open the return air damper when warming up the building in the morning. Switch No. 2 closes the by-pass under the reheating coils, and switch No. 3 cuts out the tempered air thermostat so that the by-pass will close and steam remain on these coils. In this way all the heating surface will be kept in full operation

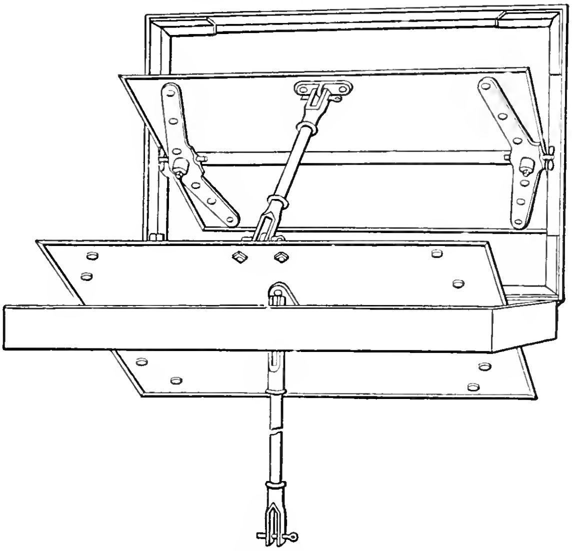


FIG. 20. MIXING DAMPERS.

since no air can short-circuit by the coils and the heating of the building will be greatly hastened. All switches have an *on* and *off* position and are placed in the boiler or engine room or wherever most accessible.

INSTALLATION AND SPECIFICATIONS

The *installation* of an automatic temperature control system is usually made by the manufacturer of the particular system selected, and should only be attempted by experienced work-

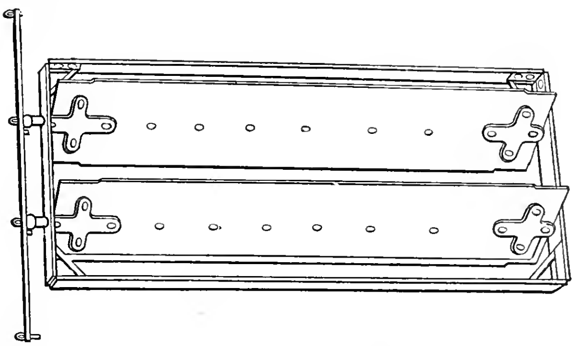


FIG. 21. LOUVERED DAMPERS.

men who are specialists in this field. The piping system must be run true and straight with due allowance for drainage, and drip cocks should be installed at all low points. Galvanized

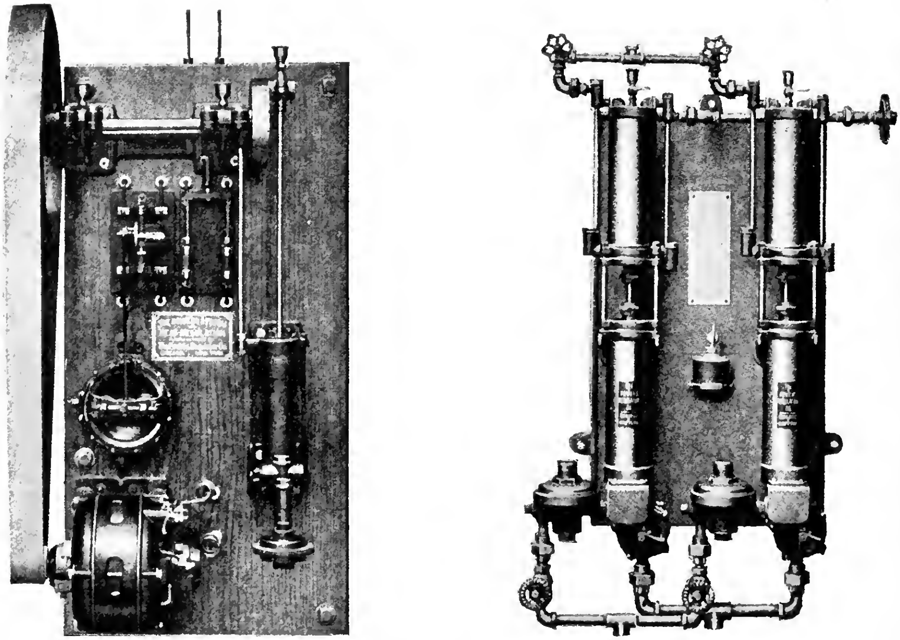


Fig. 22. Air Compressors for Powers System.

2-inch Electric Compressor. 3-inch Hydraulic Compressors in Duplicate.

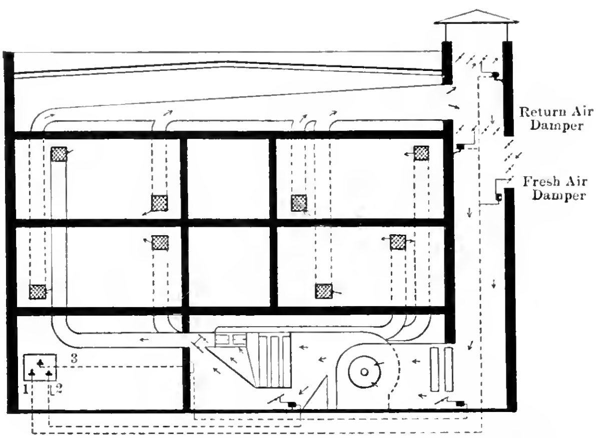


Fig. 23. Johnson Remote Control System for Dampers.

piping or flexible metal tubing is used throughout, and all main lines are $\frac{3}{8}''$ or $\frac{1}{2}''$ in diameter. Branches are made $\frac{1}{4}''$ and lines to separate thermostats are made $\frac{1}{8}''$ in diameter. The use of cast brass fittings is recommended on these small lines. Where flexible connections are required armored lead tubing is used.

Due consideration must be given to the placing of the suction intake to the compressor, and the proper elimination of oil and dirt from all compressed-air lines provided for.

Specifications. Specifications for such a system are seldom prepared in detail. The *U. S. Treasury Dept.* imposes certain requirements in this class of work which follow, and are taken from the specifications for the new *Jersey City* post-office building:

"*Automatic Temperature Controlling Apparatus.* Furnish and install a complete system of automatic temperature controlling apparatus for both the direct radiation and the fan blast heating and tempering coils.

"All rooms and corridors are to be maintained at 70° except those noted in schedule at 65°.

"*Thermostats* must be of approved design and must have a neat index and pointer, and those in rooms and corridors must have a neat and reliable mercurial thermometer attached and riveted or otherwise substantially secured in such a way as to prevent its easy removal. Thermostats in rooms and corridors to be located about 5′-6″ above the floor, and in such position as will best serve to give the average temperature desired. When more than one thermostat is placed in a room they must be so located that a natural division of the room may be made by future partitions. Thermostats controlling direct radiators are to be ornamental in design and finished to harmonize with the hardware trimmings in each room. Each to be neatly and firmly attached to the wall or columns and to be capable of adjustment at least 5° each way from the temperature desired in the room where placed. Thermostats must be of such design as to prevent their being thrown out of adjustment by the occupants of the room.

(A schedule of rooms, thermostats, and radiators is here given. There are 43 room thermostats.)

"All *automatically controlled valves* on the direct radiators are to be heavy pattern "radiator" valves of first class and approved manufacture, rough body, finished trimmings, nickel plated all over, provided with ground joint union connections and elastic discs. They must be provided with suitable diaphragm attachment of sufficient size to insure steam tight valves with 10 pounds air pressure and 20 pounds steam pressure. All valve stems on direct radiators are to be set vertical except on radiators in basement. The weight of body, union connection and construction of stuffing-box to be the same as specified for radiator valves with hand control. See requirements for radiator valves as given under Heating Specifications.

"For the *control of the fan blast heating and tempering coils,* thermostatically operated valves will be placed on both the steam and return connections to each individual heater section of the sizes shown on plans. These valves to be of the same quality as above specified for radiators, except that they need not be nickel-plated and a separate brass ground joint union may be placed on each side of the thermostatic valves.

" For the control of these valves and the by-pass dampers under the heating and tempering coils the following thermostats must be used: One thermostat at the cold air inlet set at 40° to operate the steam and return valves on the outside section of the tempering coil; one thermostat in tempered air space at outlet of air washer set at 49° to operate the steam and return valves on the two remaining sections of the tempering coils; one thermostat at same location set at 51° to operate the by-pass damper under the tempering coil.

"At the fan outlet one 2-point thermostat having the first point set at 71° to operate the steam and return valves on the outside section of heating coils, and the second point set at 73° to operate the steam and return valves on the remaining section of heating coils. Also one thermostat at fan outlet set at 75° to operate the by-pass damper under the coils.

"All *duct thermostats* must be capable of adjustment to 15° or more one each side of the point at which they are set, and must be properly shielded from radiant heat. Those controlling

steam, return, and radiator valves to be positive acting and those controlling dampers to be of graduated type.

" Furnish and install near the power switchboard, where shown on plans, one *air compressor* of proper size, driven by an electric motor of approved make and wound for the proper speed at 500 volts direct current. The compressor may be direct connected to its motor, or may be driven through the medium of a noiseless chain drive, or spur or worm gear running in an oil bath. Air compressor and motor to be mounted on a cast iron plate and suitable concrete foundation.

"A galvanized steel *storage tank* of not less than 4 cubic feet capacity is to be supplied and erected with proper connections to air mains. The connection between pressure tank and system is to be provided with a positive-acting pressure-reducing valve to maintain a practically constant pressure of 15 lb. on the diaphragm valves. A polished brass pressure gage with a 5″ dial must be installed on this line at the point indicated.

" Motor and compressor to be painted in color as directed.

" Automatic temperature controlling devices not known to this office must be described in detail and satisfactory evidence submitted that similar systems are in entirely successful operation.

" The *motor* for driving the compressor shall be supplied with an automatic starting rheostat that will start the motor when the pressure in the air storage tank drops to 15 lb. and stop

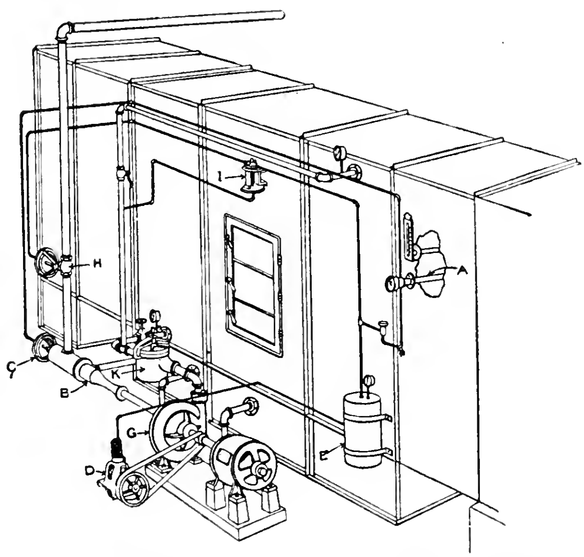


FIG. 24. GENERAL ARRANGEMENT OF THE CARRIER SYSTEM FOR HUMIDITY CONTROL.

same when pressure reaches 35 lb. Rheostat must have a suitable overload and no voltage release.

" Rheostat, switch, etc., and, so far as practical, all electrical devices controlling this apparatus, are to be mounted on the power switchboard hereinafter specified. Electrical connections are to be made as hereinafter specified.

"All *air distributing lines* shall be of ample size to carry the amount of air required for each line, and must be run in a neat and workmanlike manner concealed behind plaster, etc., except in the basement, where lines are to be run exposed on the ceiling. All air lines are to be of galvanized wrought iron pipe, with tees, ells, couplings, etc., of galvanized cast iron, or of cast brass."

HUMIDITY CONTROL SYSTEMS

Humidifying by Means of Air Washers. The control of the relative humidity in any building is most positively and satisfactorily accomplished by the use of a *humidifying type* of air washer in which sufficient spray nozzles of proper design are installed to saturate the air leaving the washer at the *proper saturation temperature*, so that when this air is finally warmed to room temperature it will have the desired relative humidity.

The general arrangement of the humidity control system employed by the *Carrier Air Conditioning Co.* is indicated in Fig. 24. The stem of a graduated thermostat shown at *A* is placed in the air passage just beyond the eliminators, so that it is exposed to the temperature of the air leaving the washer, and its expansion or contraction is caused entirely by this temperature, and the variation due to its expansion is made to regulate this temperature. The water heater of the ejector type shown at *B* is placed in the suction line of the pump. The heater operates like a barometric condenser, so that the temperature of the spray water is varied by varying the amount of steam furnished by the ejector. The diaphragm steam valve

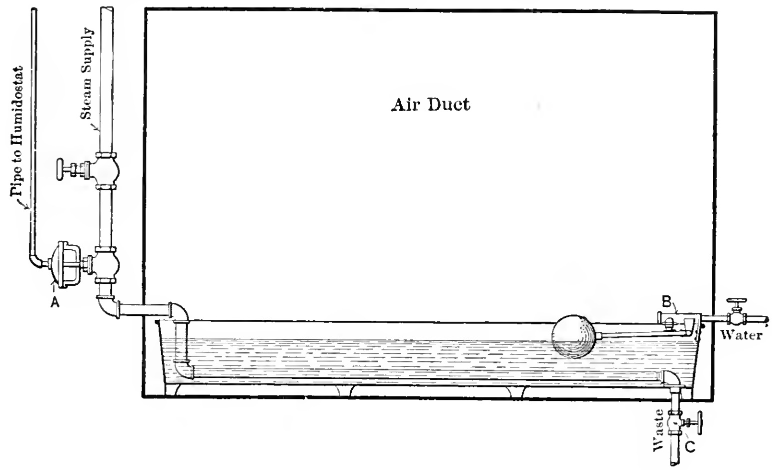


FIG. 25. CROSS-SECTION OF AIR DUCT WITH HUMIDIFIER.
(*Johnson Service Co.*)

shown at *C* is placed in the steam line which supplies the water heater. This valve is operated by compressed air from the graduated thermostat *A*.

The air compressor shown at *D* furnishes compressed air at about 15 lb. pressure to the storage tank *E*. The compressor is driven by the same motor that drives the spray water circulating pump *G*.

The reverse acting diaphragm valve shown at *H* is normally closed, but is opened by compressed air from tank *E*, passing through the safety valve *I*.

The pot strainer shown at *K* is for the purpose of catching any scale and dirt which may be brought from the steam lines.

This method of control is extremely sensitive, as any variation in the air temperature passing over the stem of the graduated thermostat produces a change in the air pressure on the diaphragm steam valve, causing the valve to take a new position, thereby producing a new water temperature. In only a few seconds this water is sprayed into the air, affecting its temperature, giving to it more or less heat in accordance with the requirements of the thermostat. This air, in about one second, passes over the stem of the thermostat, imparting to it the change in temperature, thereby completing the cycle.

Humidifying by Jets, Sprays, and Evaporating Pans. Such an expensive apparatus as an air washer cannot always be installed, and hence recourse must be had to such devices as will accomplish the result in some other, though usually less satisfactory, manner. Three methods are available, and will give fair results when under the control of humidostats which are capable of regulating within 4 per cent of the desired percentage of relative humidity.

If *live steam*, free from oil or other impurity, is available, it may be blown directly into the ducts supplying air to the rooms to be humidified. A diaphragm valve is placed on the steam-supply line, and an air-line connection made to the humidostat which can then control the amount of steam supplied as required.

If a live steam supply is not available or is unsatisfactory a *hot water spray* may be discharged through suitable nozzles into the air current. A standard diaphragm valve and humidostat will readily control the hot-water supply, although suitable provision must be made for draining away any surplus water not absorbed by the air.

The remaining method consists in placing an *evaporating pan* (Fig. 25) with steam coil in the main air duct. This pan is equipped with a ball cock supply, *B*, and the steam line to coil is fitted with a diaphragm valve, *A*, operated by air through the humidostat located in the room at some favorable point about 5′ from the floor.

The *humidostat* under the influence of change in humidity upon its sensitive element operates similarly to the thermostat to supply or release air for operating the diaphragm control valve.

CHAPTER XVIII

EXHAUST STEAM HEATING*

Economy of Heating by Exhaust Steam. The economy of using exhaust steam for heating purposes is quite apparent when it is realized that usually not more than 7 per cent of the heat above 32° F., supplied the average non-condensing engine appears as work in the steam cylinder. A large proportion of the exhaust may be used for heating, drying, evaporators, etc., and the power generated by the engines become a by-product or, from another point of view, the heating is practically free of cost.

As shown in the chapter devoted to thermodynamics,

$$1 \text{ h.p.} = \frac{33{,}000 \times 60}{778} \text{ or } 2546 \text{ B.t.u. per hour.}$$

Let S = weight of dry steam used by the engine per i.h.p. hour.

$$\frac{2546}{S} = \text{B.t.u. utilized by the engine per lb. of steam.}$$

$$H_1 = \text{total heat of steam at initial pressure per lb.}$$

$$\frac{2546}{S} \;/\; H_1 \text{ or } \frac{2546}{SH_1} = \text{fractional part of heat supplied that is utilized.}$$

Example. A simple non-condensing engine uses 30 lb. steam per i.h.p. hour with an initial pressure of 100 lb. gage and H_1 = 1190.7 B.t.u. Then $\frac{2546}{30 \times 1190.7} = 0.071$ part of heat supplied that is utilized by the engine. There still remains 1190.7 × (1−0.071) or 1106.2 B.t.u. above water at 32° F. per lb. supplied. If this exhaust is passed through the radiation of a heating system all of this heat can be utilized except that which is contained in the condensed returns at the temperature at which the condensate leaves the radiation. Assuming that the pressure carried in the radiators is practically atmospheric the temperature of the condensate leaving the radiation will be approximately 200°; therefore 1106.2 − (200–32) or 938.2 B.t.u. per lb. will be available for heating purposes or 938.2/1190.7 = 78 per cent of the total heat supplied above water at 32° F.

Assuming that live steam at a pressure of 100 lb. gage is throttled down to atmospheric pressure and used in the radiation there would be 1190.7 − (200–32) or 1022.7 B.t.u. available per lb. supplied.

As was previously shown in Chapter I the heat content for a throttling process remains constant. The temperature of the condensate leaving, the radiation is assumed as 200° F. as before. Therefore heating by exhaust steam utilizes 938.2/1022.7 or 92 per cent of the amount of the heat available for heating purposes when live steam is used direct. Practically a deduction of approximately 10 per cent must be made from the above figure to allow for heating the feed water and various losses in order to obtain the net amount available at the radiation.

Exhaust Steam Available. The exhaust steam from a non-condensing plant available for heating purposes may be approximately estimated as follows:

It is assumed that an estimate of or the actual daily load curve of the plant is at hand and the engine load curve determined. See example, Volume II.

In a simple non-condensing plant the direct acting boiler feed pump will require approximately 4 per cent of the total steam generated by the boilers. If automatic stokers are used

* The reader is also referred to the Volume on small Power Plants where much of the apparatus which is very briefly described in this chapter is more fully discussed.

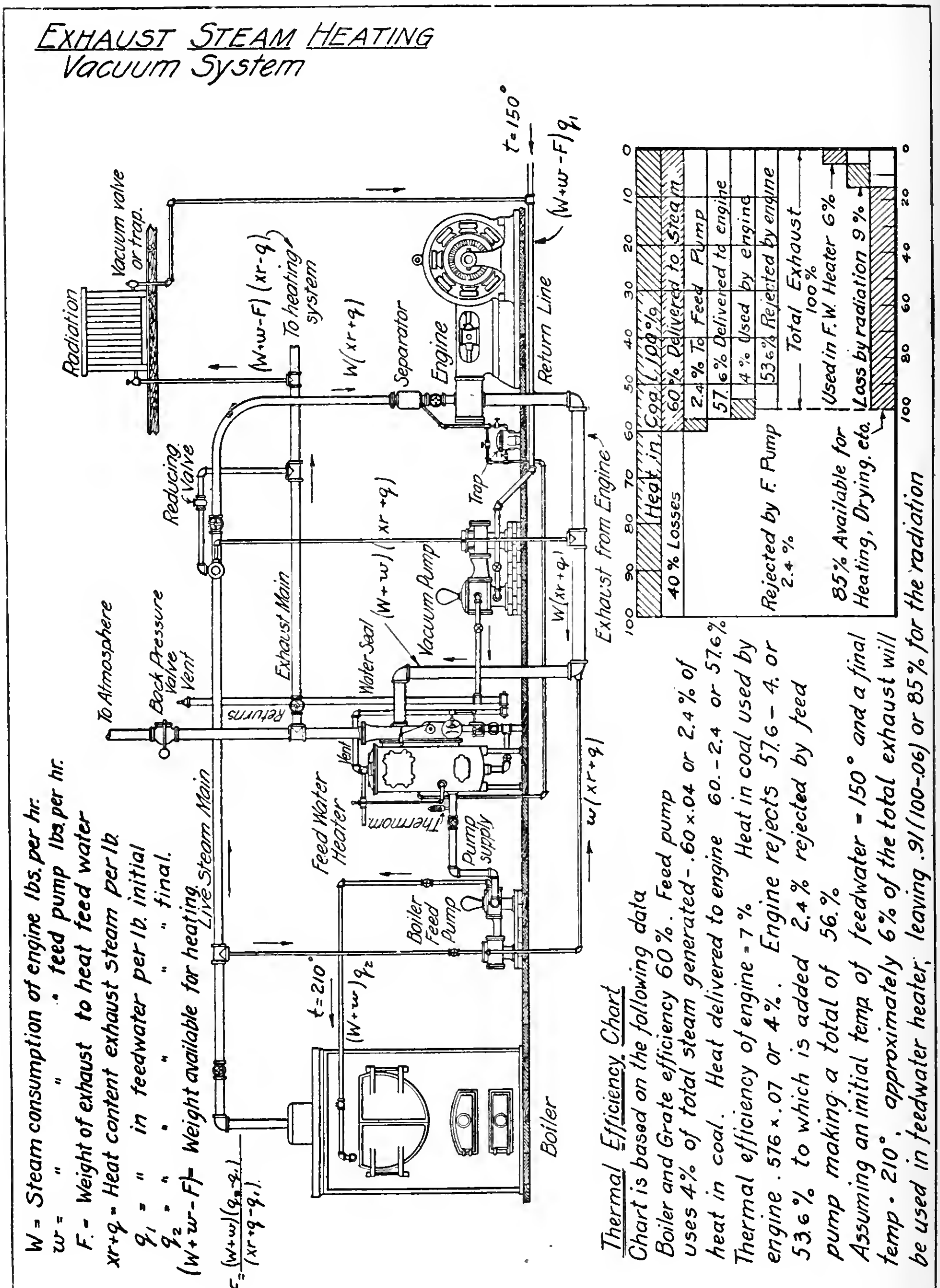

EXHAUST STEAM HEATING
Vacuum System
W = Steam consumption of engine lbs. per hr.
w = " " feed pump lbs. per hr.
F = Weight of exhaust to heat feed water
xr+q = Heat content exhaust steam per lb.
q1 = " in feedwater per lb. initial
q2 = " " " final.
(W+w−F) = Weight available for heating.
F = (W+w)(q2−q1)/(xr+q−q1)
Radiation
Vacuum valve or trap.
t=150°
(W+w−F)q1
(W+w−F)(xr−q) To heating system
W(xr+q)
Separator
Engine
Return Line
Reducing Valve
Trap
Exhaust from Engine
To Atmosphere
Back Pressure Valve
Vent
Returns
Exhaust Main
Water Seal
(W+w)(xr+q)
Vacuum Pump
W(xr+q)
Live Steam Main
Feed Water Heater
Vent
Thermom
Pump supply
Boiler Feed Pump
w(xr+q)
t=210°
(W+w)q2
Boiler
Heat in Coal 100%
40% Losses
60% Delivered to Steam
2.4% To Feed Pump
57.6% Delivered to engine
4% Used by engine
53.6% Rejected by engine
Rejected by F. Pump 2.4%
Total Exhaust 100%
Used in F.W. Heater 6%
Loss by radiation 9%
85% Available for Heating, Drying, etc.
Thermal Efficiency Chart
Chart is based on the following data
Boiler and Grate efficiency 60%. Feed pump uses 4% of total steam generated − .60 × .04 or 2.4% of heat in coal. Heat delivered to engine 60. − 2.4 or 57.6%
Thermal efficiency of engine = 7%. Heat in coal used by engine .576 × .07 or 4%. Engine rejects 57.6 − 4. or 53.6% to which is added 2.4% rejected by feed pump making a total of 56.%
Assuming an initial temp. of feedwater = 150° and a final temp. = 210°, approximately 6% of the total exhaust will be used in feedwater heater, leaving .91/(100−.06) or 85% for the radiation

FIG. 1.

approximately 2 per cent and for forced and induced draft 5 per cent of the total amount of steam generated will be used respectively by these auxiliaries. The feed water heater will take approximately 17 per cent of the weight of exhaust for heating the feed water from 50° F. to 210° F., and 6 per cent for heating from 150° F., assumed temperature of returns, to 210° F., see "Feed Water Heaters." With this data and the economy curves of the engines an estimate may be made of the weight of exhaust steam available.

Let S = total weight of steam used by engines and auxiliaries per hour.

X = fractional part of the weight of exhaust used to heat the feed water.

XS = weight of exhaust condensed in feed water heater per hour.

then $(1 - X)\,S$ = weight of exhaust steam available for heating or drying purposes per hour, and deducting 9 per cent for unavoidable losses leaves $(0.91 - X)\,S$ lb. available.

When the condensation from the radiation is returned to the heater at an average temperature of 150° F., $X = 0.06$ and the weight available is 0.85 S per hour. It is seen that about

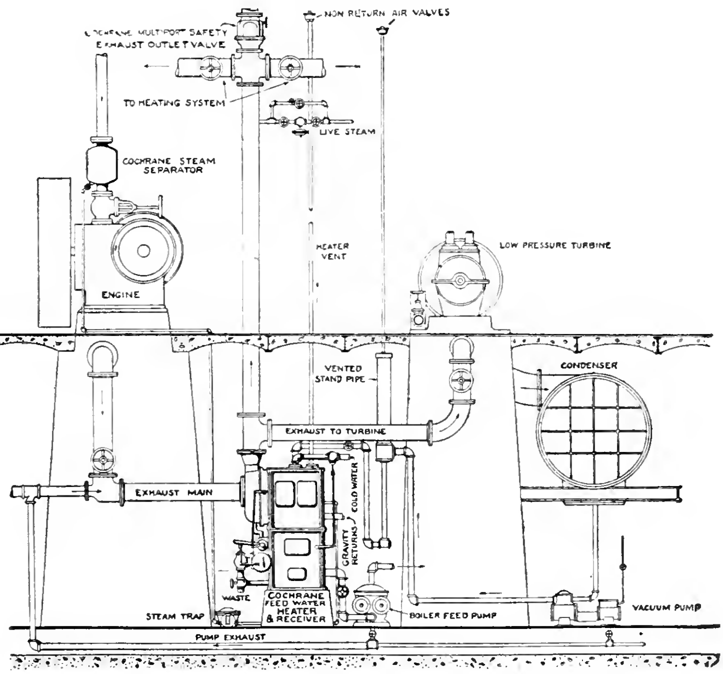


Fig. 2. Arrangement of Exhaust Steam Heating System.

85 per cent of the weight of the exhaust from the engines is available for heating in the non-condensing plant with no auxiliaries except the feed pump when the condensation from the radiation is all returned to the feed water heater. Allowing for condensation in the exhaust mains, etc., we may expect approximately this proportion of the engine exhaust to reach the radiation.

Example. Thus for a plant equipped with simple high speed engines operating at one-half load "S" from the water rate curves, Section 2 is 33½ lb. for an initial pressure of 100 lb. gage. The weight of exhaust steam available is therefore approximately 0.85 × 33.5 or 28.5 lb. per I. H. P. hour. Allowing 0.25 lb. per sq. ft. of direct radiation gives 28.5/0.25 or 114 sq. ft. per I. H. P. of engines.

Assuming a ratio of 1.55 I. H. P. to 1 K. W. output of generator, the weight of exhaust available per K. W. output is 1.55 × 28.5 or 44.2 lbs. The radiation supplied per K. W. will be 44.2/0.25 or 177 sq. ft.

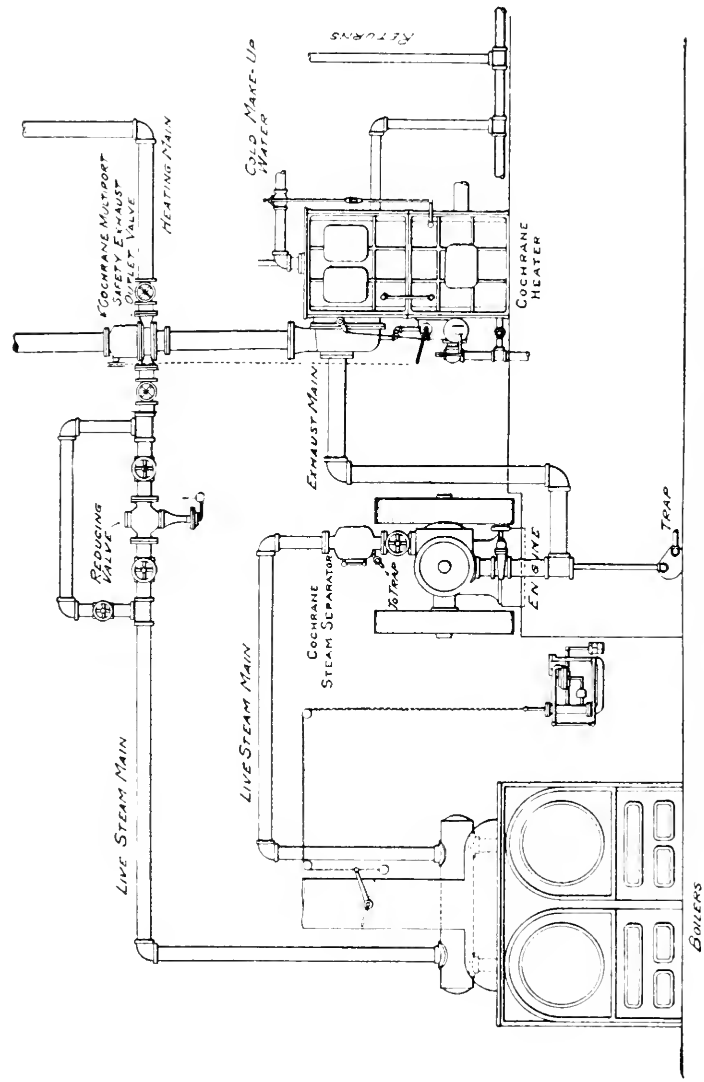


Fig. 3. Arrangement of Exhaust Steam Heating System

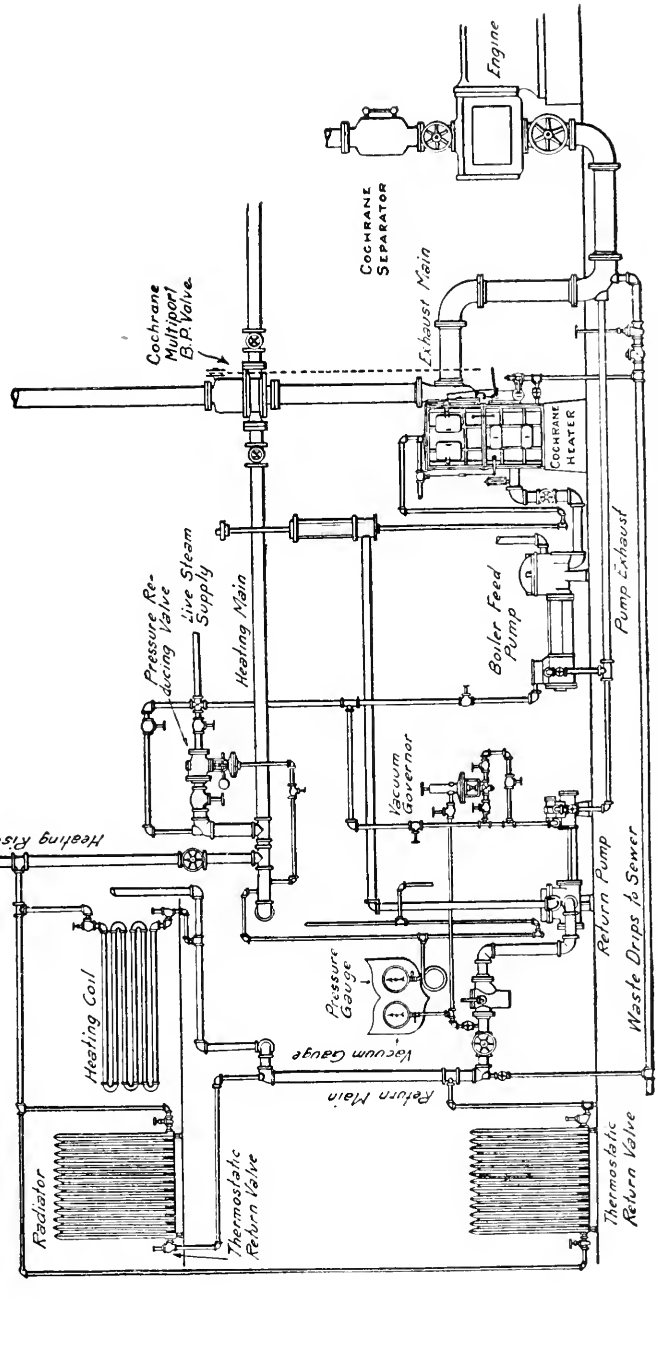


FIG. 4. VACUUM SYSTEM AS APPLIED TO EXHAUST STEAM HEATING.

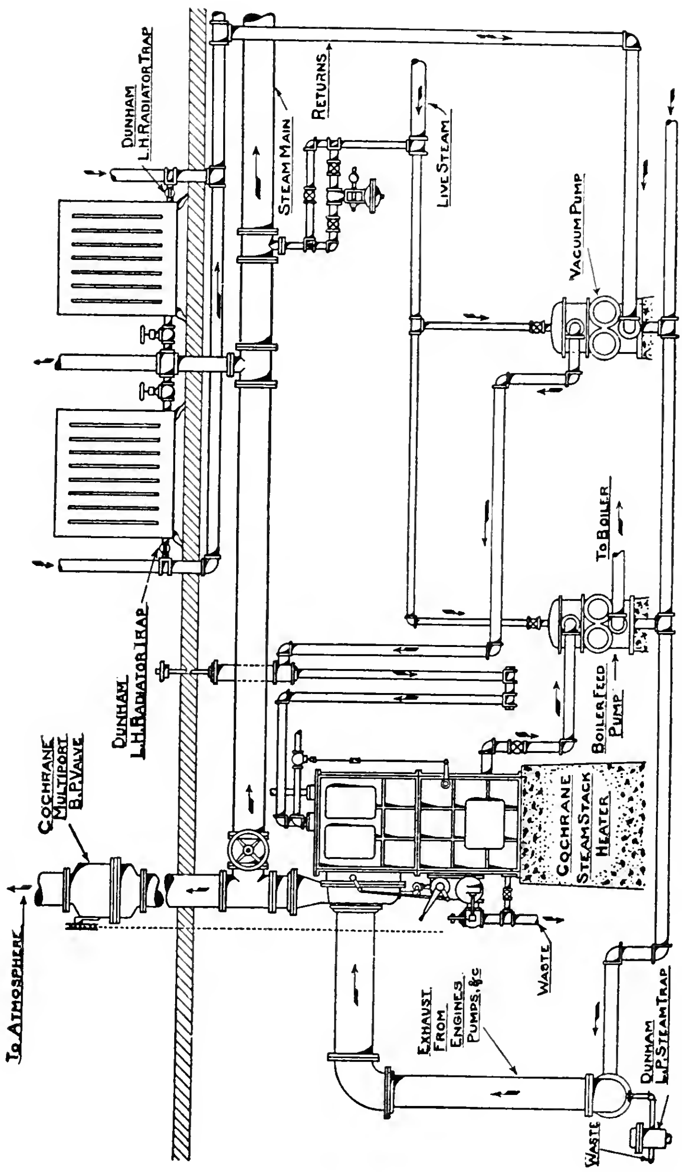


Fig. 5. Vacuum System as Applied to Exhaust Steam Heating.

Exhaust Steam Heating Systems. In exhaust steam heating it is essential, in order to obtain good circulation and prevent high back pressure on the engines, to install a vacuum system. In this system each radiator and each section of a hot blast heater is equipped with a vacuum valve or trap on the return end which allows the air and condensate to pass through but no

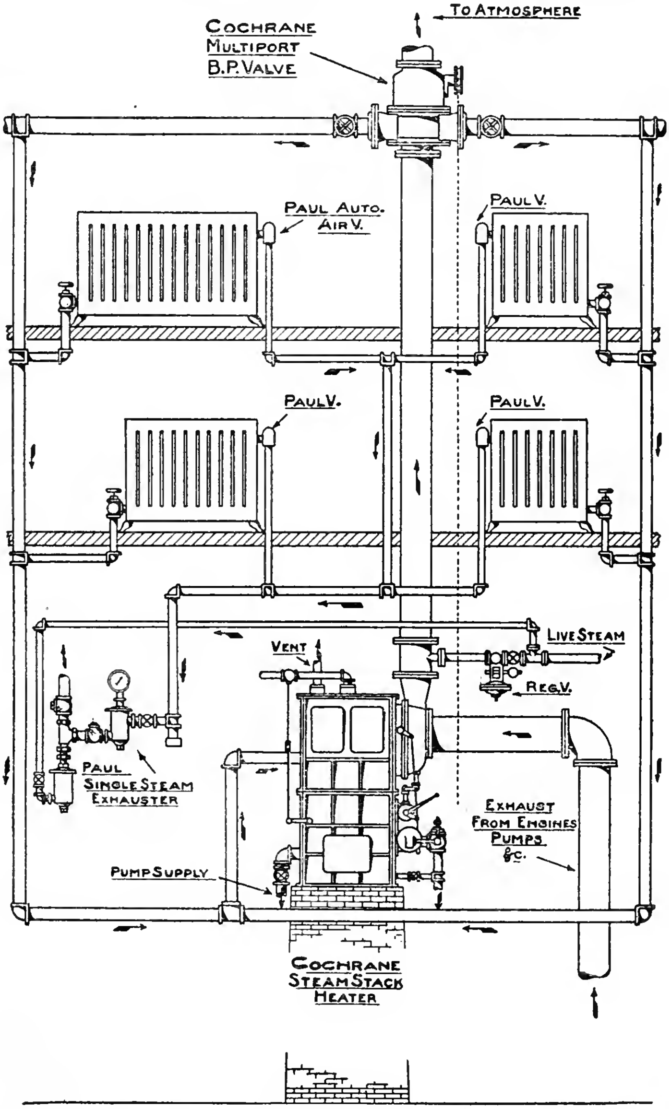


FIG. 6. AIR LINE VACUUM SYSTEM AS APPLIED TO EXHAUST STEAM HEATING.

steam. See chapter on "Direct Steam Heating." This trap is generally of the thermostatic type. A vacuum pump designed to handle both the air and water is attached to the main return. This pump maintains several inches of vacuum in the return so that the vacuum valves discharge against practically no back pressure.

The automatic and positive removal of the air and condensate insure the circulation of the steam at practically atmospheric pressure when the steam mains and branches have been properly designed. In this connection a pressure loss of about 1 oz. or $\frac{1}{16}$ lb. per 100 ft. of pipe, when not exceeded, gives good results. See "Friction Pressure Loss Diagram" in chapter on "Direct Steam Heating." A live steam by-pass, equipped with a reducing valve, connects the exhaust main with the high pressure main. The reducing valve automatically opens and admits live steam reduced to the pressure of the exhaust whenever the amount of exhaust is insufficient to supply the radiation and during periods when the engine is shut down.

The various parts which are included in an exhaust steam heating system are shown by Fig. 1.

The make-up water for the system may be conveniently introduced into the vacuum pump suction in the form of a spray. This serves to condense any vapor that may have formed from the condensate when the vacuum valves discharge into the lower pressure maintained in the return lines.

Several arrangements for vacuum exhaust steam heating systems are shown by Figs. 3 to 6.*

Size of Vacuum Pump Required. The following table by the *Warren Webster Co.* may be used in determining the size of vacuum pump necessary. To determine the size of pump required the following empirical formula is used:

Square feet of direct radiation + (Number of units × 100) = F. Choose the nearest size corresponding to the value of F given in the table. The steam cylinders are proportioned on a basis of 80 lb. pressure and for lower pressures the steam cylinder must be proportioned accordingly.

Example. Required the size of pump for 5000 sq. ft. of direct radiation in 150 radiators. 5000 + (150 × 100) = 20,000. Use a 5″ x 6″ x 10″ pump.

TABLE 1

SIZES AND CAPACITIES OF VACUUM PUMPS

Size	Steam	Exhaust	Suction	Discharge	F	Floor Space	List Price
	Inches	Inches	Inches	Inches			
4 x 5	⅜	...	....	....	6830	....	....
4 x 4 x 6	⅜	½	2½	2	7270	11 x 34	$125.00
4 x 4 x 8	⅜	½	2½	2	8000	11 x 34	150.00
5 x 5	...	...	....	....	10680	....	....
4 x 5 x 6	¾	½	3	2½	11353	13 x 36	160.00
4 x 5 x 8	⅜	½	3	2½	12500	13 x 38	170.00
4½ x 5½ x 8	½	¾	3	....	15125	13 x 38	180.00
6 x 5	...	...	....	....	15990	....	....
6 x 7	...	...	....	....	17215	....	....
4½ x 6 x 8	½	¾	3	2½	18000	13 x 38	190.00
5 x 6 x 10	¾	1	4	3	19390	18 x 50	225.00
7 x 7¼ x 10	1	1¼	5	4	28256	18 x 52	330.00
6 x 7½ x 12	¾	1	5	4	30605	19 x 54	315.00
7 x 8 x 16	1	¼	5	4	34470	18 x 52	355.00
6 x 8 x 12	¾	1	5	4	36620	19 x 54	340.00
9 x 10	...	...	....	....	43627	....	....
8 x 9¼ x 12	1	1½	6	5	48957	20 x 58	440.00
8 x 10 x 12	1	1½	6	5	57220	20 x 58	450.00
10 x 14	...	...	....	....	60250	....	....
10 x 12 x 12	1¼	1½	6	5	82997	20 x 62	550.00
10 x 16	...	...	....	....	30713	....	....
14 x 16	...	...	....	....	122500	....	....
12 x 14 x 20	1½	2	10	8	133000	....	1075.00
16 x 16	...	...	....	....	161270	....	....
14 x 16 x 20	2	2½	12	10	173720	....	1375.00
16 x 18 x 20	2½	3	12	10	219870	....	1600.00
18 x 24	...	...	....	....	233640	....	....
18 x 20 x 20	3	3½	14	12	271440	....	1920.00

Steam and Electric Load Curves in Large Buildings. The relation between the steam requirements for heating and electric loads in large buildings is a question upon which very little

* From catalog of *Harrison Safety Boiler Works.*

accurate data are available. The committee on heating of the *National Electric Light Association* collected a number of curves typical of different types of building and presented some of them in the report. This collection of curves is reproduced herewith. ("Power," July, 1913.)

The charts include curves showing the steam requirements of the building and curves showing the steam which would be required for the electricity actually consumed, based on a factor of 60 lb. per kw.-hr. In the case of some buildings where pumps or steam-driven elevators are installed, there is an additional curve showing the high-pressure steam required for these pumps, and a curve showing the total steam which would be available for exhaust from both the pumps and electric units, if they were installed. The curves show the actual steam required, and in the case of the curve showing steam available for exhaust, a slight deduction should be made on account of the condensation removed by steam and oil separators.

Fig. 7–1 shows the steam and electrical requirements of a department store in *Detroit* on a typical winter day. This store is a brick building of five stories, having cubical contents of 1,545,000 cu. ft. and connected steam radiation of over 10,000 sq. ft. There are also installed in the store a duplex steam pump, 18 x 12 in., and a house pump, 12¼ x 10 in. The store is open for business from 8 A.M. until 5:30 P.M.

Fig. 7–2 represents the steam and electrical requirements during a summer day of a large department store in *Chicago*. This building is eight stories in height and the cubical contents are approximately 7,800,000 cu. ft. In addition to radiation for both direct and indirect steam heating, there is required steam for 16 elevators and for boiler-feed pumps. Fig. 7–3 shows the steam and electrical requirements in the same department store during the holiday trade, in ordinary winter weather, and Fig. 7–4 during severe winter weather.

Fig. 7–5 shows the steam and electrical requirements in an office building in *Detroit*. This is an eight-story building of brick construction having cubical contents of 565,000 cu. ft. and connected steam radiation of 6,600 sq. ft.

Fig. 7–6 shows the steam and electrical load in an office building in *New York*. This is a twenty-four story building, having cubical contents of 6,500,000 cu. ft. and having 63,140 sq. ft. of radiation connected. This is a composite curve averaging 28 typical winter days.

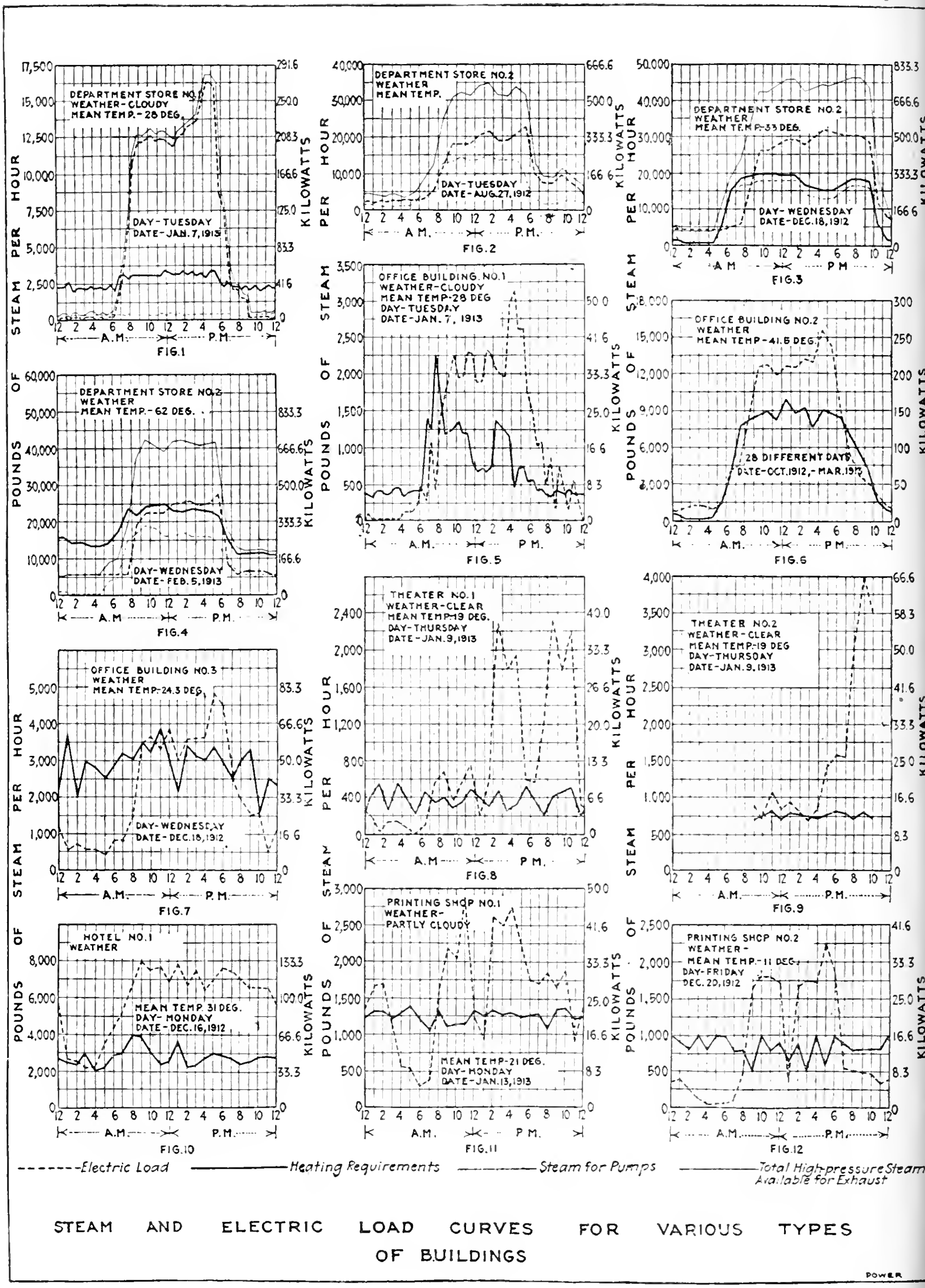
Fig. 7–7 shows a large office building in *St. Paul*. This is a steel and brick building, 16 stories high, having cubical contents of 1,907,800 cu. ft. and connected steam radiation of 19,500 sq. ft. This diagram shows conditions on a typical winter day in *St. Paul*. Steam is kept on all night in cold weather, thus giving a more even steam line than for some of the buildings.

Fig. 7–8 represents the steam and electrical requirements in a theater in *Detroit*. This is a five-story building of brick construction, having cubical contents of 710,000 cu. ft. The steam for the office portion of this building, however, is not shown in the diagram, simply the steam and electrical requirements in the theater. This curve shows that there are two performances each day in the theater. Fig. 7–9 is a diagram for another *Detroit* theater. This building is brick, with a stone front, having cubical contents of 750,000 cu. ft. The steam for the office building is not always shown in the diagram. Fig. 7–10 shows the steam and electrical requirements of a large hotel in *St. Paul*, on a typical winter day. This is a 12-story building, of tile and steel construction, having cubical contents of 2,308,270 cu. ft. The heating load in this building is fairly uniform and the steam required for the electrical load is not only variable, but runs far in excess of the steam used for heating.

Fig. 7–11 shows the steam and electrical requirements in a printing shop in *Detroit*. This is a five-story brick building, having cubical contents of 315,000 cu. ft. and a connected steam radiation of 4,008 sq. ft. The hours for this shop are 7:30 A.M. to 5.15 P.M. There is also a night shift with regular hours.

Fig. 7–12 represents a printing shop in *St. Paul*. It is a four-story brick building, having cubical contents of 695,320 cu. ft. and connected radiation of 9,529 sq. ft. This is a publishing and printing house, and the hours for the night shift are irregular. It will be noticed that the electrical load is used principally in the daytime.

Comments on the charts follow to the effect that even in the winter time, when the heat-



STEAM AND ELECTRIC LOAD CURVES FOR VARIOUS TYPES OF BUILDINGS

FIG. 7.

ing and power requirements more nearly approach each other, there is still a wide variation in the two curves, some showing that at certain periods of the day the demand for steam for heating is in excess of the demand for electricity, while at other periods of the day the demand for electricity is greater than the demand for heating. In view of the fact that the steam demand for heating disappears altogether in the summer, and in the winter is not coincident with the demand for electricity, the utilization of exhaust steam for heating in isolated plants, in the opinion of the committee, is not nearly as great a factor in economy as has sometimes been claimed. The question is not whether heating can be done economically with exhaust steam, but whether the entire plant can be operated economically. In the most modern buildings the heating load constitutes a comparatively small part of the total requirements of the building.

Condensing Plants. When condensing water is available at the cost of pumping, and there is an opportunity at the same time of utilizing at least a fair proportion of the exhaust steam from the engines or turbines for heating or process work, it becomes necessary to decide between the installation of a non-condensing and a condensing plant. A compound engine running condensing will use approximately 33 per cent less steam than a simple non-condensing machine, 20 per cent less steam than a non-condensing compound engine, all conditions being equally favorable. This, however, does not mean that the saving in fuel will be equal to this amount, as the pumps required for handling the condensing water are now an added tax on the plant as a power consumer and an increased fixed charge also due to the additional condenser equipment and larger size engine cylinders or turbines required.

It frequently happens that the location of the plant is such that water is costly to obtain even when water-saving devices, as cooling towers, etc., are installed to conserve it. It should be borne in mind, however, that pumps which consume power must be used in any event, and what may on the face appear a large saving may in reality not pay much of a dividend on the extra investment involved, particularly in small plants. The condensing auxiliaries in a medium size plant consume approximately 8 to 12 per cent of the steam used by the engine when condensing water is available at the plant with practically no pumping expense, so that the net saving in fuel will amount to approximately 11 per cent, against which must be figured the difference in fixed charges between a non-condensing equipment of equal capacity and the condensing equipment being considered.

The following example will serve to illustrate the principles involved in making a comparison between a condensing and non-condensing plant when steam is also required for heating or process work. Assuming a 500 horsepower non-condensing compound engine operating 300 days in the year, 10 hours per day, and a heating load to be taken care of equal to 50 per cent of the power load for 6 months in the year, steam consumption of engine and feed pump 24 + 4 per cent or 25 lb. per I. H. P. hour. This plant requires, if run at normal load, an evaporation of:

$500 \times 25 \times 10 \times 300 = 37{,}500{,}000$ lb. of water per year.

A 500 horsepower compound condensing engine using 18 + 10 per cent (for operating auxiliaries) or 20 lb. per I. H. P. hour requires an evaporation of:

$500 \times 20 \times 10 \times 300 = 30{,}000{,}000$ lb. per year.

The radiation will require an additional amount equal to 50 per cent of the amount used by the non-condensing plant for 6 months or

$$0.50 \times 0.50 \times 37{,}500{,}000 = 9{,}375{,}000 \text{ lb.}$$

The total evaporation for the condensing plant plus the heating load is therefore 39,375,000 lb. per year, which is seen to be greater than required by the non-condensing plant. In general it may be said that when the heating load for approximately 6 months of the year is equal to roughly 25 per cent or more of the power load it is more economical to install a non-condensing plant, or if approximately 15 per cent of the exhaust may be utilized continuously, as for process work, the same statement holds good.

COMBINED SYSTEMS

Condensing and Non-Condensing Plants. In many cases where the power load is large in comparison with the heating load a combined system will be found to be the most economical arrangement. When all of the exhaust can be utilized during the heating season it will pay to shut down the condenser during the season when heat is required and use the exhaust from the engine for heating. Unless the engine is overloaded the cut off may be increased to admit more steam to make up for the power lost by loss of the vacuum.

If only a portion of the exhaust can be used, say 50 per cent, and the load is divided between several units one or more of which may be run condensing, then the exhaust from the other unit may be used for heating.

If the compound condensing engine is installed steam may be drawn from the receiver for heating after having been expanded down to approximately 20 lb. gage in the first cylinder and the remainder passed through the low pressure cylinder and further expanded to the condenser pressure. This requires a special regulator which controls the cut off of the low pressure cylinder from the pressure in the receiver.

The relation between the type of heating system and the machine used may be expressed as follows: If the heating plant requires steam in amount equal to or slightly exceeding that furnished by the engines, at a pressure of 4 to 8 lb. per sq. in. above atmospheric, a single-cylinder engine is advisable, because this pressure corresponds to the final pressure for a diagram showing the best economic operation of such an engine. If at times a little more steam is required for heating, it may be taken direct from the boiler, and brought down to the required pressure by a reducer. If, however, the quantity of steam required for heating is considerably less than that delivered by the engines, or is at a pressure of 8 to 35 lb. per sq. in. above atmospheric, a compound engine should be used, with the steam taken at the receiver and constant pressure maintained by varying the admission to the low-pressure cylinder.

This relationship exists also in the case of turbines, the high pressure turbine corresponding to a single-cylinder engine, and the two-stage turbine to a compound engine, the steam in the latter case being taken at the point where its pressure corresponds to that in the heating system.

When steam going to the low-pressure cylinder is taken to the heating system, the work that would have been developed in it has to be made up by the high-pressure cylinder, and if much steam is regularly taken in order to insure normal working of the engine, the volume of the low-pressure cylinder must be decreased. If too much steam were taken, the low-pressure cylinder would receive none, and would give no work; the engine would then work as a single-cylinder engine with back-pressure, driving the low-pressure cylinder at no load with a corresponding reduction in the efficiency of the machine. On the other hand, if too little steam is taken from the receiver then too much load is thrown on the low-pressure cylinder. There are therefore certain upper and lower limits of admission to the low-pressure cylinder which must be maintained by some method of automatic regulation.

Reciprocating Engines and Turbines. A combination of a reciprocating engine and low pressure turbine may also be used in this connection. The engine exhausts at approximately atmospheric pressure into a receiver as an open feed water heater from which the supply to the heating system and the condensing turbine may be drawn.

The above arrangements have been frequently used in plants where it has been desired to increase the plant capacity and still make use of the original equipment.

A variation of the above may be made when two turbines are installed, part of the exhaust from the first being utilized for heating or process work and the remainder passed through the second or low pressure condensing turbine. The low pressure turbine may be operated on high pressure steam exhausting to atmospheric if the condenser should fail, giving emergency possibilities not possessed by the engine if carried out as a compound as previously mentioned.

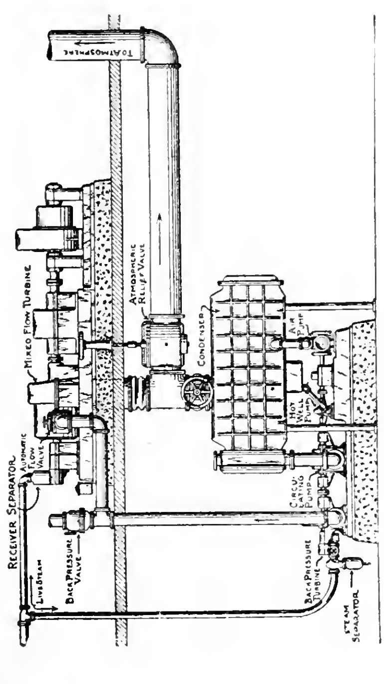


FIG. 8-1

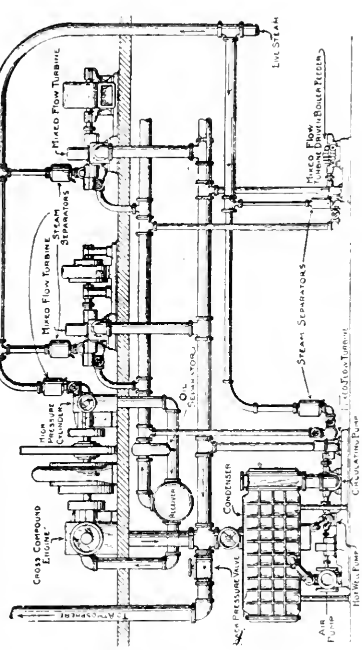


FIG. 8-2.

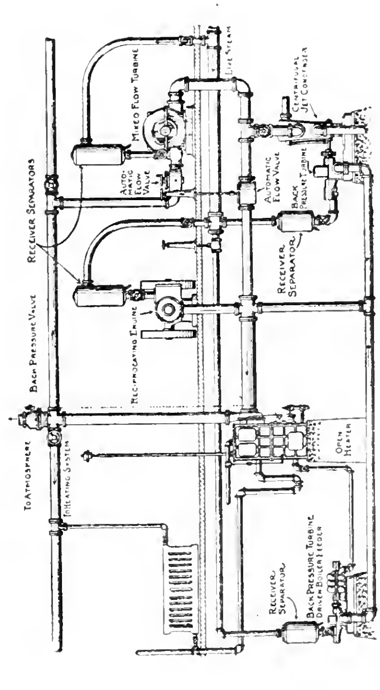


FIG. 8-3

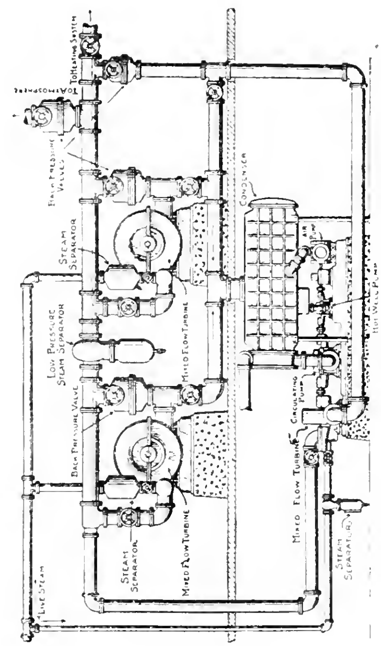


FIG. 8-4.

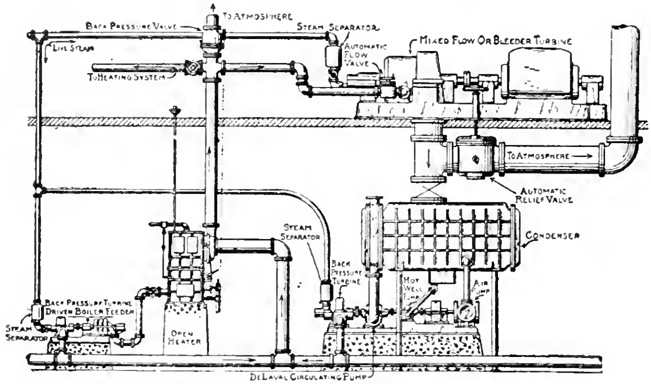
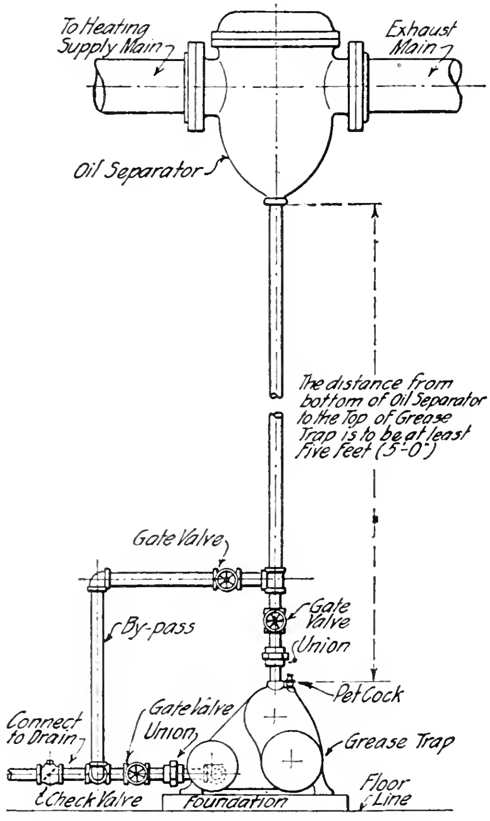


FIG. 8–5.—(*Continued.*)



Typical of Drip Connections from Oil Separator thru Grease Trap to Drain As applied to the Vacuum System of Steam Circulation.

FIG. 9.

Mixed flow turbines are obtainable in which both high and low steam may be used combining the features above mentioned in one machine.

The various arrangements previously described are shown by figures taken from an article by *Geo. H. Gibson*, published in *The Journal of the Engineers' Society of Pennsylvania*, May, 1913.

The following notes refer to the figures indicated.

Fig. 8–1. Back-pressure turbine driving auxiliaries in condensing steam turbine plant. The condenser auxiliary turbine can exhaust either to atmosphere or to an intermediate stage

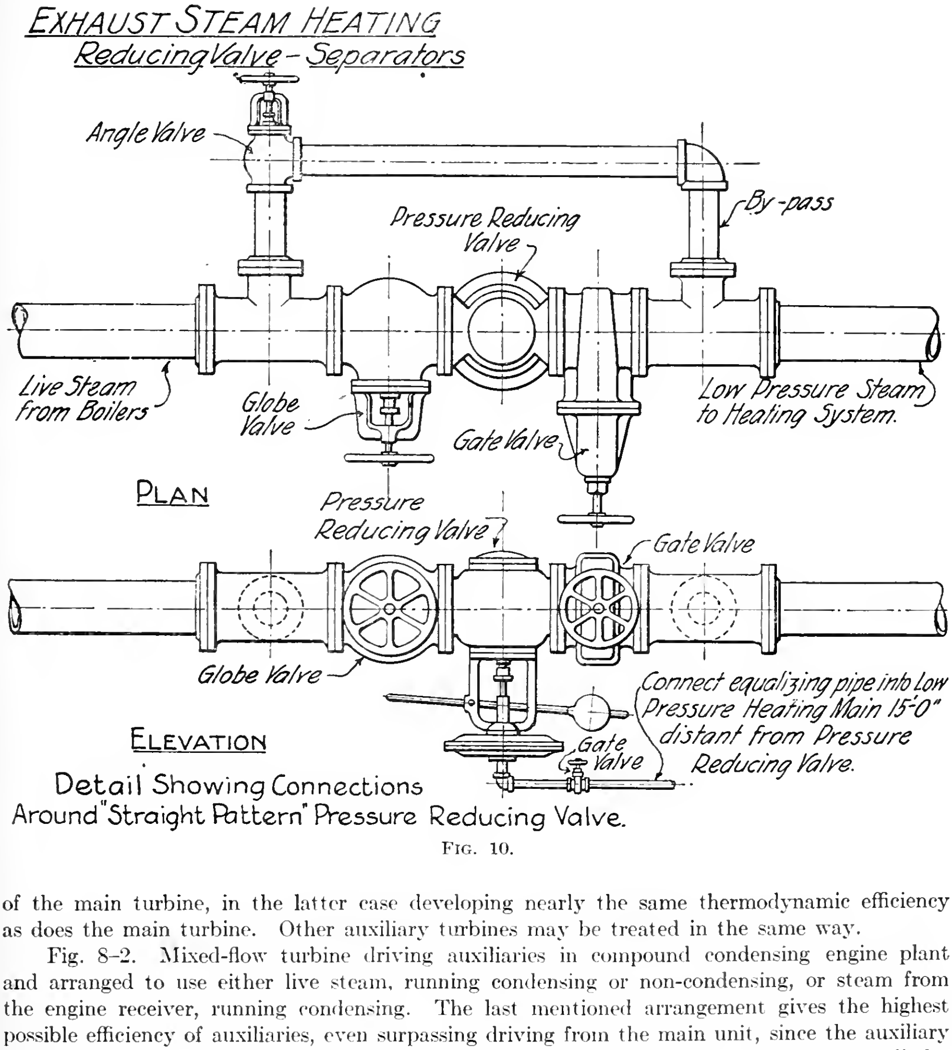


Fig. 10.

of the main turbine, in the latter case developing nearly the same thermodynamic efficiency as does the main turbine. Other auxiliary turbines may be treated in the same way.

Fig. 8–2. Mixed-flow turbine driving auxiliaries in compound condensing engine plant and arranged to use either live steam, running condensing or non-condensing, or steam from the engine receiver, running condensing. The last mentioned arrangement gives the highest possible efficiency of auxiliaries, even surpassing driving from the main unit, since the auxiliary turbines running on low-pressure steam are much more efficient than is the low-pressure cylinder of the engine.

Fig. 8–3. Mixed-flow turbine, for increasing the capacity and improving the economy of a reciprocating engine operated in connection with an exhaust steam-heating system. The

turbine may receive either surplus exhaust steam, discharging to condenser, or when heating requirements are severe, live steam, discharging into the exhaust steam-heating system. It may also be operated upon live steam, condensing, when engine is shut down, in which case boiler feed water will be heated by exhaust from auxiliaries.

Fig. 8–4. Two mixed-flow steam turbines, arranged to form combined exhaust steam-heating and condensing system of maximum economy. Either one or both of the main turbines may be run on live steam, exhausting to heating system, or to atmosphere, or to condenser. When steam is not required for heating, one of the units can run on low-pressure steam exhausting to condenser. The auxiliary turbine can be used on either the high-pressure or the low-pressure side of the circuit, accordingly as the proportion of steam required for heating varies.

Fig. 8–5. Condensing steam turbine plant, in which the exhaust steam from the back-pressure auxiliary turbines, non-condensing engines, etc., is used first to heat boiler feed water, the surplus steam passing to the heating system or to the intermediate stage of the main turbine, giving the auxiliaries and engines nearly as high thermodynamic efficiency as has the main unit. Where the amount of steam required by the heating system is more than the auxiliaries can supply, the main turbine can be of the *bleeder* type.

CHAPTER XIX

CENTRAL STATION OR DISTRICT HEATING

SYSTEMS IN USE

The distribution of either steam or hot water from a single central plant for supplying a city district (Fig. 20) or a group of buildings is a more or less common method of heating where conditions are favorable. The area served should be as compact as possible, the connected load should be large and steady, and the heating plant should preferably be located at a low point near the center of the district or the group served in order to equalize the length of the distributing mains. A distributing radius of one mile or even more is not at all unusual, even though *the deterioration and loss of heat from the underground mains* are the most serious objections to the installation of a central heating system.

The original district heating plant was installed by the late *Birdsall Holly* in the city of Lockport, N. Y., in 1877, and supplied *live steam* through about one mile of underground piping to five stores, seven residences and two churches. The water of condensation was returned to the plant and re-evaporated. Hot water was first used for central heating about ten years later.

Advantages of a Central Heating System. The advantages of such a system are found in the elimination of the smoke nuisance, since it is entirely possible to burn the cheaper grades of fuel both efficiently and smokelessly in a central plant with constant attendance. The economy of the plant depends on whether it is a *simple* heating plant or a *combined* heating and power plant; in the latter case the steam for heating is practically a by-product if the exhaust is sufficient in amount to supply the heating requirements. According to *Mr. B. T. Gifford*, in commercial plants of this latter type: " The revenue from heat sales will more than pay the coal bills after deducting interest, maintenance and depreciation on the heating investment from the heat income. It can be seen, therefore, that the cost of electricity per kilowatt-hour ought to be less in the combination plant. It does not increase the labor cost, as the same crew can handle both services."

The advantages of the service to the consumer involve several factors, such as cleanliness, convenience, readiness, etc., although for the same space heated it will usually cost more than the coal required for an isolated heating plant.

Practically all of the commercially successful plants are of the combination type supplying power, heat and light. See the discussion of the central heating system for West Chester, Penn., given later.

Comparison of Steam and Water Systems. A brief comparison of the steam and water systems in use to-day indicates that the *installation cost* of the steam system is less for the plant equipment, although the distributing system runs about the same for either. The water system is much more flexible in responding to the heating requirements, and hence the *operation cost* is generally less. This difference is not so marked if all steam radiators are thermostatically controlled.

The *regulation* of the outgoing water temperature so as to just supply the heat loss corresponding to the outside temperature and wind conditions is a decided advantage in favor of hot water, in addition to its economy, since it eliminates the necessity for thermostatic control

in the buildings heated. Moreover, the *line losses* from hot-water mains at an average temperature of 160° F. are less per square foot of surface than from steam mains at 220° F. or higher.

Hot-water mains are readily adapted to *undulating ground conditions* and may follow the contour of same, provided service branches are taken off at the high points to relieve the air. Steam and return mains must be uniformly graded, and if relayed at any point, suitable drip connections must be made through traps or seals. Single pipe steam mains with no returns may be used for undulating ground conditions, but in this case the condensation from each building must be wasted to sewer or drain.

It is also possible to use exhaust steam at *low pressure*, that is, atmospheric or less, from either condensing or non-condensing units for supplying the hot-water heaters. In the former case the heaters are placed between the engines and condensers, and a saving effected in the amount of condenser water required. A *back pressure*, above the atmosphere, of several pounds is always necessary in a steam system.

Hot-water systems are subject to *pressure limitations* when operating with cast-iron radiators, which are only tested to 90 lb. cold-water pressure. The maximum static head in the system should not exceed 100 ft. on the lowest radiator, or $\frac{60.5}{144} \times 100 = 42$ lb. No such pressures are necessary in the steam system.

The determination of *service charges* is more easily made with the steam system, and is based on condensation meter readings, rather than taken as a flat rate, and based on square feet of radiation installed.

For a very complete discussion of methods and systems in use for arriving at proper service charges for both steam and water heating from a central station, see "District Heating," by *S. Morgan Bushnell* and *Fred B. Orr*.

CENTRAL STEAM-HEATING SYSTEMS

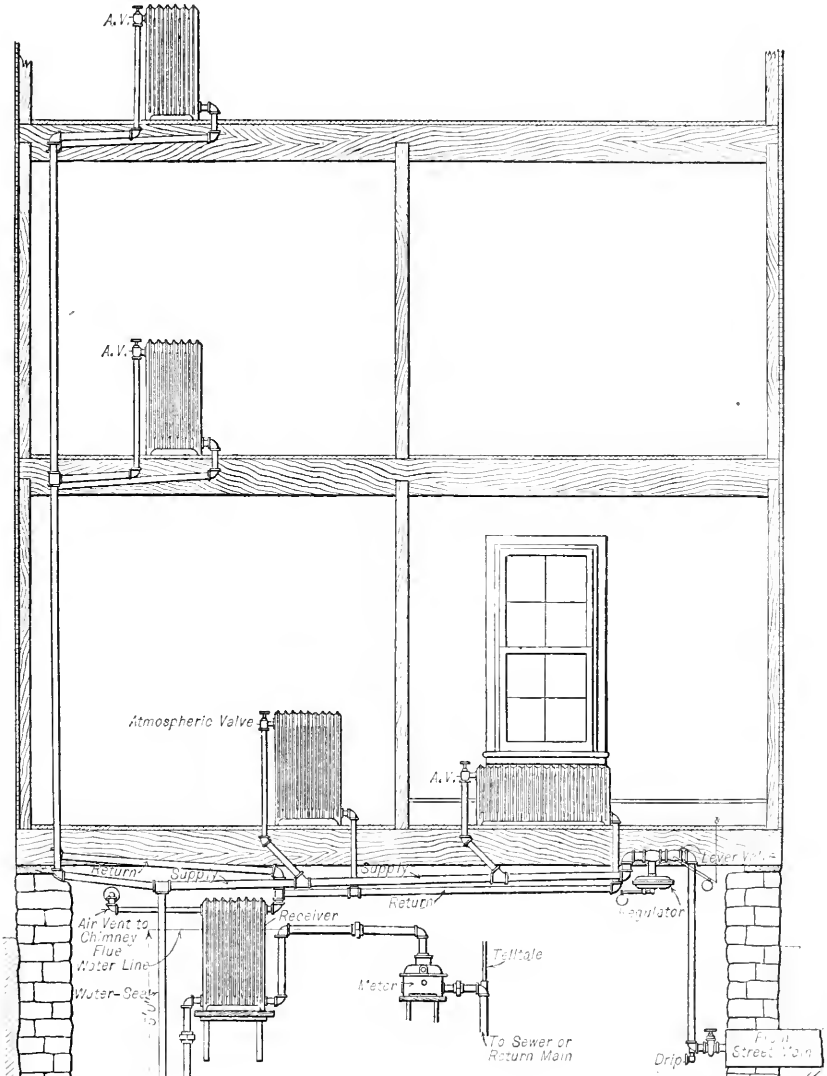
Operating Conditions. Steam may be distributed at *high or low pressure*, depending on whether live or exhaust steam is to be used in the mains. In the former case steam is used at 10-lb. pressure or above, while in the latter case the pressure usually ranges from 2 to 5 lb. at the station, and a pump is used on the returns to create a vacuum and facilitate the return of the condensation so that the back pressure on the engines may be kept as low as possible. By discharging the condensation to the sewer it is possible to avoid the pressure loss in the returns and eliminate the vacuum or return pumps.

The later development of central steam-heating systems had resulted, generally, in combining them with light and power plants in order to make use of the *exhaust* steam from the engines and pumps so that *live* steam need only be used to supplement the exhaust. It has also been found advisable in many cases of district heating not to attempt to return the water of condensation from the distant buildings, but to discharge same directly to the sewer after it has been cooled as much as possible by circulating it through an "economizing coil," located in the basement of the building.

Essential Features of a Steam System. The essential features of a central steam-heating system include: (1) The building equipment with radiation, secondary piping, cooling or economizing coil and meters if the service is to be metered. The economizing coil is not needed if all condensation is returned to the plant. (2) The distributing mains provided with expansion joints, anchors and suitable covering, all carried in the necessary conduit or tunnels. (3) The steam-generating plant or central station, which must provide either live or exhaust steam or both. Pressure-reducing valves will be necessary at the plant as well as back-pressure valves and oil separators, if exhaust steam is used in the heating mains. The usual feed-water heaters and boiler-feed pumps must be provided, as well as suitable vacuum pumps in case the condensation is to be returned as in a low-pressure system.

(1) Building Equipment

Radiation and Piping. The buildings to be heated must be supplied with sufficient *radiation* to offset the heat loss when operating with steam at from 1 to 2 lb. gage pressure, unless an "atmospheric system" is installed and usually designed to operate at from 5 to 8 ounces gage pres-



Interior Steam Piping Atmospheric System.

FIG. 1. ATMOSPHERIC SYSTEM.

sure. It is possible to heat the buildings with water radiation using a steam coil heater in the basement for heating the circulating water, in which case the radiation must be proportioned as for water heating.

The *amount of radiation* is determined as already explained for direct steam and water heating in the chapters on "Heat Transmission of Building Construction and Direct Radiators." In case an "atmospheric system" is installed the radiation as calculated above must be increased about 15 per cent and the economy coil and trap on the returns are omitted. Air valves are not required on this system but must be installed on all "low-pressure" systems.

The *building piping* is run as for any standard low-pressure system with either basement or overhead mains, and the outside service line connects into top of same at the highest point (Fig. 1). This service line is equipped with a stop valve just inside the building wall and may be operated by a chain attached to a weighted lever, and run to the floor above. A drip cock is placed just beyond this valve at the point where service line rises to supply building main.

All steam piping should be covered, but returns may be left bare in case condensate is to be discharged to sewer, or if the system is to operate at "atmospheric" pressure.

The return lines should be brought together, properly sealed, and trapped to discharge into the building return if sent back to plant, or through an "economizing coil" or "receiver" if discharged to sewer.

Atmospheric System for District Heating. The building piping for an atmospheric system using a receiver instead of an "economizing coil" on the returns is shown in Fig. 1. The *pressure regulator* or reducing valve maintains from 5 to 8 ounces in the building mains. A compound pressure and vacuum gage (not shown) should be connected to both sides of the regulator. All piping is designed for 1 ounce loss per 100 ft. of run.

The radiators are connected on a two-pipe system, and are of the water type, with top and bottom tappings. They are controlled by a $\frac{3}{4}''$ graduated supply valve with ports of such size as to admit just the required amount of steam to heat each radiator, which is made 15 per cent greater in area than for ordinary low-pressure service. A special union elbow is used on the return end of each radiator and acts as a seal against the escape of steam. The radiator condensation is discharged into a receiver, and the steam main is dripped into the same receiver through a water seal at least 5′–0″ deep, as shown.

The receiver has an air vent connected to a near-by chimney, or running to the outer air. This vent line should run at least 20 feet vertically; to prevent flooding and the escape of steam.

Economizing Coil and Trap. The system shown above may be operated as a low-pressure steam system, provided the receiver, which is there shown vented to the atmosphere, is replaced by an economizing coil (Fig. 2) and a suitable trap installed between the returns and the coil.

This discharge *trap* on the return main must operate practically at atmospheric pressure, and, if of the float type, there should be a 3-inch water seal over the seat when the outlet valve closes.

The trap is installed as shown in Fig. 2 and must be placed below the lowest steam radiator, and at least six inches above the highest outlet at top of "economizing coil," if one is used. These traps must be proportioned to the amount of radiation served as indicated in Table 1.

TABLE 1

CAPACITIES OF EMPIRE STEAM TRAP

(*American District Steam Co.*)

Number	Inlet, Inches	Outlet, Inches	Capacity, Sq. Ft. of Radiation	Weight, Pounds
0	1	$\frac{3}{4}$	500	30
1	$1\frac{1}{4}$	1	1,200	59
2	$1\frac{1}{2}$	$1\frac{1}{4}$	1,900	72
3	2	$1\frac{1}{2}$	4,000	117

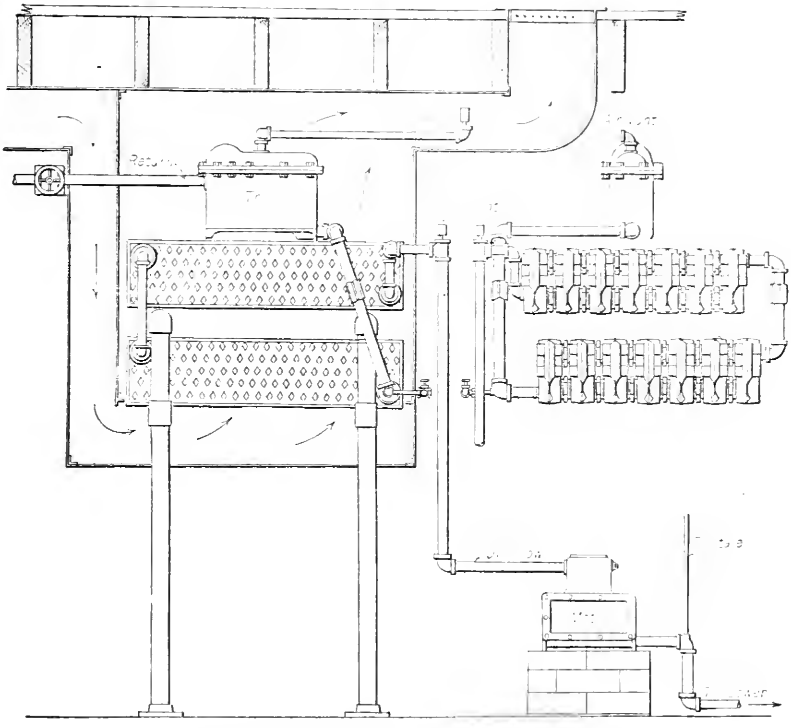


FIG. 2. ECONOMIZING COIL AND CONNECTIONS

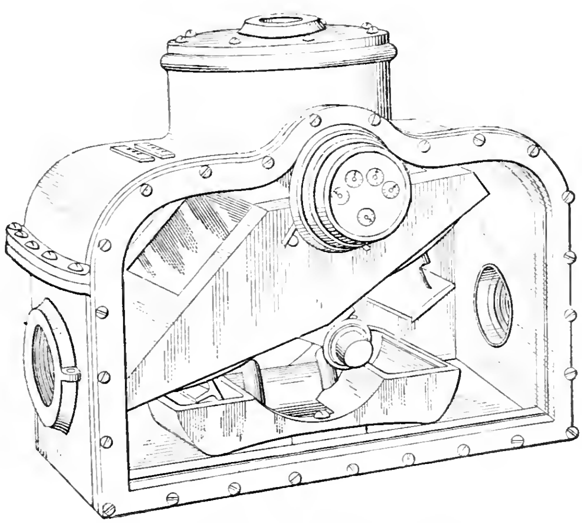


FIG. 3. SIMPLEX CONDENSATION METER.

The *economizing coil* should have a heating surface equal to approximately 20 per cent of the area of the radiation from which condensation is discharged, and is set as an indirect radiator with suitable air supply and discharge ducts. Common proportions are 1 sq. in. of area in cold air duct and 1¼ sq. in. of hot-air duct for each sq. ft. of coil surface. The method of setting these coils is also shown in Fig. 2, and it will be noted that connections are so made as to keep the coil always flooded with water. In case steam is shut off, the coil must be drained through

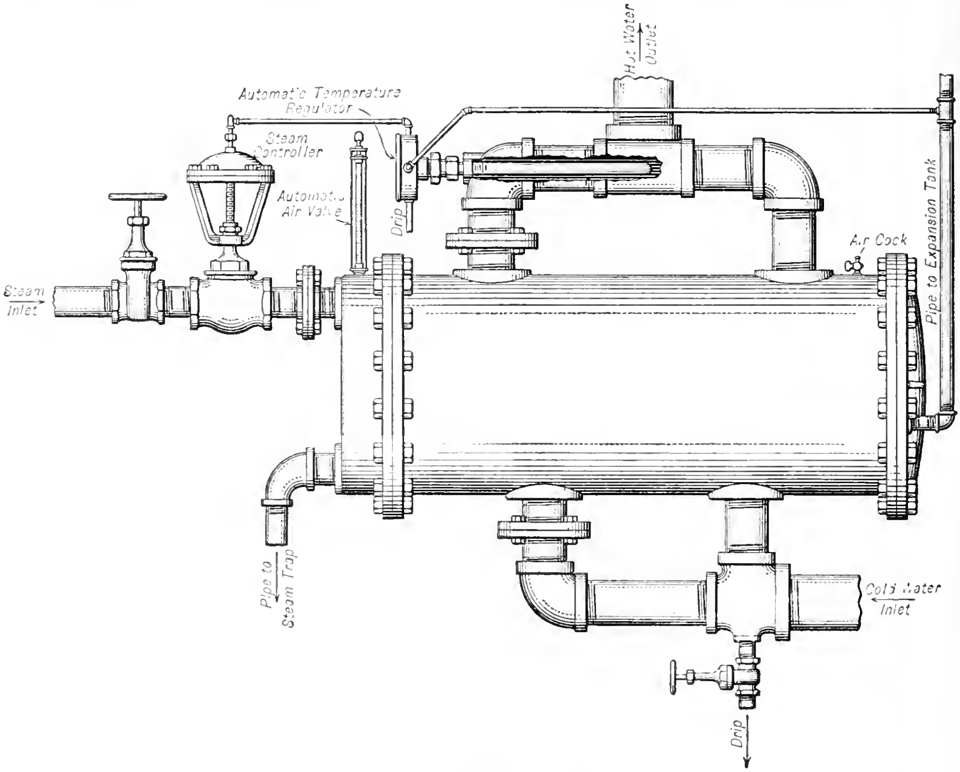


FIG. 4. HOT-WATER HEATER.

the meter as shown. Both trap and "economy coil" must be equipped with automatic air valves for venting all air pockets.

Cast-iron indirect radiator sections are generally used in making up these coils. Each section has 10 sq. ft. of heating surface, is 38½″ long by 12¼″ deep, and is 3″ wide in the stack, weighing 70 lb.

Condensation Meters. The *condensation meter* must be placed below the economizing coil and receive the discharge under simple gravity head (Figs. 2, 3 and 5). Full size connections must be used, and the discharge to sewer must be short, and grade at least one inch between the meter and sewer opening. The Simplex meter made by the *American District Steam Co.* and shown in Fig. 3 is in very common use in commercial service and operates as a tilting bucket, having two equal compartments, which hold a definite weight of condensation before tilting and emptying. Each oscillation is recorded by the register, and hence the weight of steam condensed is definitely known, and may be charged for if supplied from a commercial plant at a definite rate per 1000 lb. In practice this rate runs about 50 cents per 1000 lb., varying with the total amount used. (See "District Heating," by *Bushnell* and *Orr*, for specific rate data.)

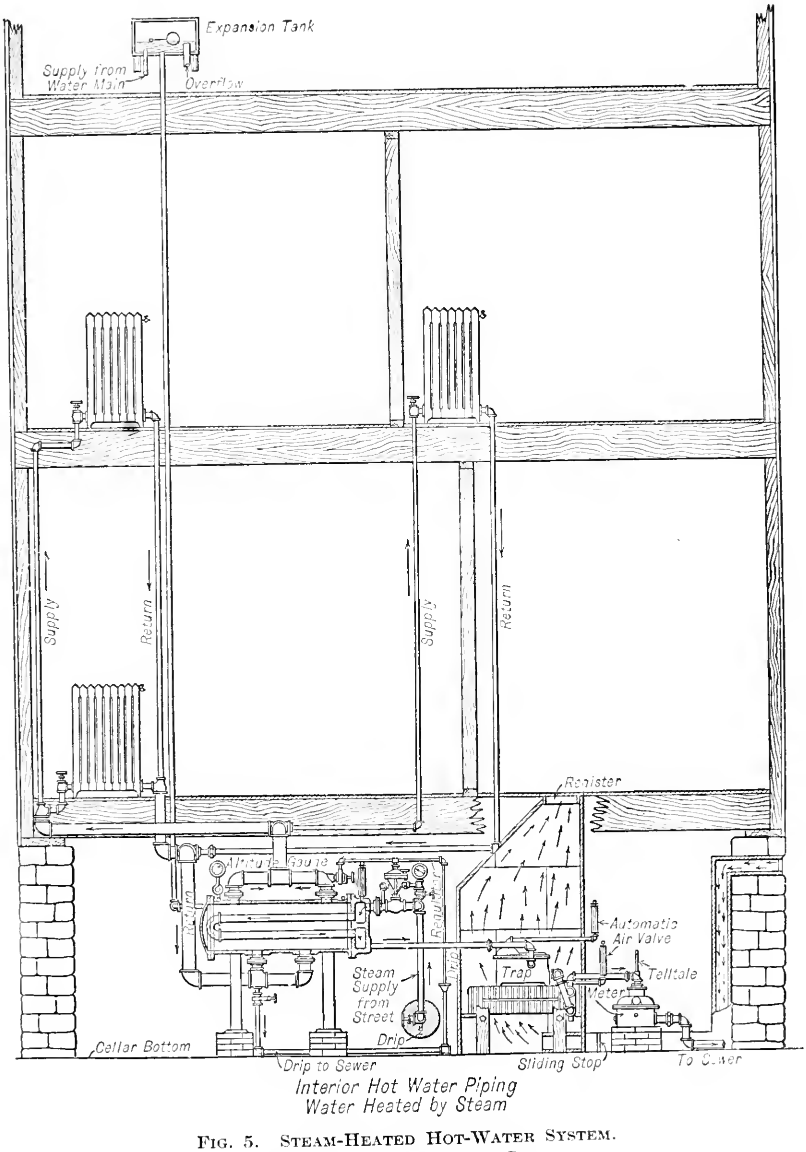


FIG. 5. STEAM-HEATED HOT-WATER SYSTEM.
American District Steam Co.

Hot-Water System Operated by Steam from Central Plant. Buildings equipped with hot-water radiation and a hot-water piping system may be heated from a central steam plant by installing a steam coil hot-water heater with temperature regulator and expansion tank (Figs. 4 and 5).

The return connections on the heater discharge line are arranged exactly as for a one-pipe, low-pressure steam-heating system, with trap, "economizing coil" and condensation meter.

Automatic Temperature Control. The final item of building equipment for central heating work should be a standard temperature control system. This is especially true where steam is supplied and paid for at a commercial rate, as it reduces waste and thereby makes it possible for *a plant of given capacity to serve more radiation.* The air mains for supplying the thermostats and diaphragm valves may be run in in the same conduit or tunnels with the heating mains, and the compressed air may be all supplied from the central plant.

(2) Distributing System

Main Piping and Fittings. The distributing system should be as short and direct as possible, and must be run in suitable underground conduit or tunnels, properly graded and thoroughly underdrained. The systems in use are either of the one- or the two-pipe type, depending on whether a return main is used for delivering the water of condensation back to the plant. In the one-pipe system the condensation is all discharged to the sewer, after recovering as much of its heat as possible by means of an "economizing coil" as already explained.

The choice between the one- and two-pipe systems depends on the cost and maintenance of the extra return piping and pumps, if necessary, as compared with the cost of water and value of the heat returned by the water. Since these costs vary with each locality no general method can be endorsed, although in city service it is usually customary to discharge the condensation to the sewer wherever good feed-water can be obtained at reasonable rates.

High- or low-pressure steam may be used in either system, but in all cases should be reduced to about 1 or 2 lb. gage before entering the building piping.

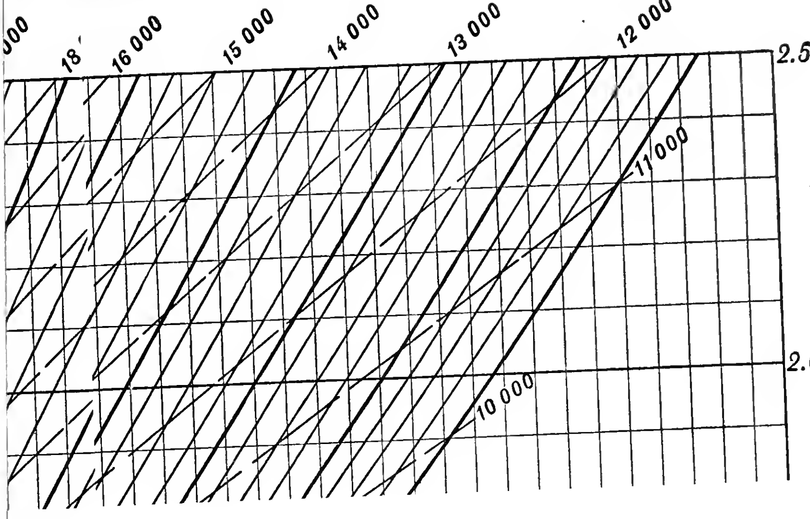
Sizes of Mains for Central Heating. The size of mains which will be required to supply a given system must be designed on a basis of maximum allowable pressure drop or friction pressure loss. If exhaust steam is used, and we must have at least one pound (better $1\frac{1}{2}$ pounds) pressure at the building, and 3 lb. is the maximum allowable at the plant against which the engines can exhaust, then our longest distributing line may have an allowable friction loss of 2 lb., but if 10 lb. is available at the plant the loss may be 9 lb.

This loss in pressure may be readily calculated for any size of pipe (diameter in inches = d), provided the length of main in feet L, the weight of steam to be supplied per minute W, and the density D, at the initial pressure are known.

The above values may be substituted in *Babcock's* formula, $p_2 - p_1 = L \times W^2 \times \frac{1}{7569\ D} \times \left(\frac{1 + \frac{3.6}{d}}{d^5}\right)$ and the friction pressure loss determined. See also the Chapter on "Water, Steam, and Air," for flow of steam.

The chart (Fig. 6) has been constructed by the use of the preceding formula in order to extend its application so as to determine quickly any one of the factors affecting the pressure loss in steam lines under a wide range of conditions. This chart has been approved by a committee of the *National District Heating Association* (see a report by Messrs. *H. A. Woodworth, J. H. Walker,* and *J. D. Hoffman*) and has been checked for ordinary ranges in velocity, but should be used with caution where velocities are high.

Example. (1) As an example for 8-in. pipe carrying 40,000 lb. steam an hour with initial pressure 125 lb. absolute, on the horizontal 8-in. line find the 40,000 division and follow vertically up to the



000
18
16 000
15 000
14 000
13 000
12 000
11 000
10 000
2.5
2.0

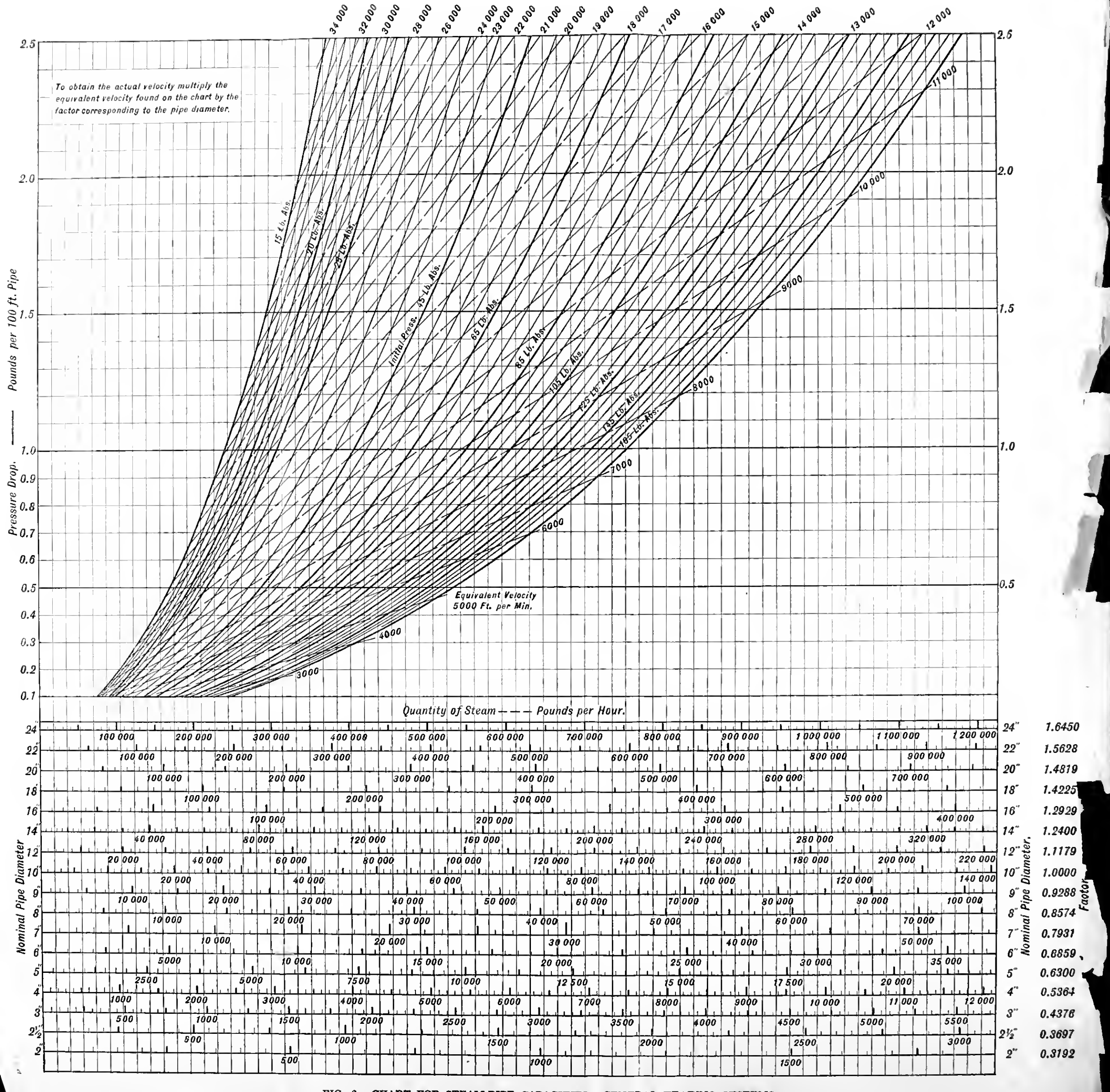


FIG. 6. CHART FOR STEAM-PIPE CAPACITIES. CENTRAL HEATING SYSTEMS.
(National District Heating Association.)

125-lb curved line. The equivalent velocity is practically 8000 ft. a minute, which, multiplied by the factor for 8-in. pipe as given at the right of the pipe diagram, 0.8574, gives 6856 ft. a minute as the actual velocity. The pressure drop per 100 ft. of pipe is found by following horizontally from the intersection on the 125-lb. curve to the right or left vertical scale to be 0.94 lb. per sq. in.

(2) If we are to supply 40,000 sq. ft. of direct steam radiation, condensing 0.25 lb. per sq. ft., or a total of 10,000 lb. per hour, with an initial pressure of 25 lb. absolute, and 0.5 lb. loss per 100 ft. is allowable, we find, by reading down from the intersection of the 0.5 lb. pressure drop line and the 25 lb. absolute curve until we cross the nominal pipe diameter line nearest to 10,000 lb. an hour, that a 7″ steam line is required.

The sizes of *return pipes* may be proportioned as for gravity flow, using the friction pressure loss curve or formulas for flow of water as given in the Chapter on "Water, Steam, and Air." In case the grade of the return main is not sufficient to give the necessary head for the required size of pipe a return pump must be used to overcome the friction pressure loss. This pump must operate as a discharge pump if it is handling hot returns, since if used on suction as a vacuum pump the hot water will flash into steam if the pressure is much reduced, and the returns are not cooled by spray or otherwise before reaching the pump.

The sizes of steam service pipes for connection between the underground main and the building may be taken from the following table. These service lines should be taken from the top of the mains at special fittings, which also serve as anchors for the main line, and any condensation in the main should be taken care of by drips and traps.

TABLE 2

CAPACITIES OF STEAM SERVICE PIPES

(Based on ¼ lb. steam condensed per sq. ft. of radiation per hr., at a pressure of 2½ lb. in the radiator.)

Length Pipe, Feet	Nominal Size Pipe, Inches						
	1½	2	2½	3	4	5	6
10	0 to 700	1,600	2,900	5,900	12,000	20,000	35,000
20	0 to 450	1,300	2,000	4,000	8,300	14,000	24,000
40	0 to 400	800	1,500	2,800	5,500	11,000	18,000
60	0 to 250	550	1,100	2,000	4,700	7,500	13,000
100	......	400	800	1,600	3,600	7,000	10,000
150	......	350	600	1,300	2,800	5,000	9,000

Mains smaller than 2½ inches diameter should not be used for underground service, even though the friction pressure loss allowable may permit a smaller size.

In plants using exhaust steam supplemented by live steam, it is sometimes possible to supply the live steam at some point distant from the station, where a large demand exists. In this way a small pipe with a large drop in pressure may be used, as boiler pressure is available for supplying this line. Since this steam does not enter the exhaust main at the station the capacity of the outlying system can be increased without increasing the back pressure on the engines.

The *amount of steam required* per square foot of direct radiation per season varies from 500 to 900 lb. per season. The committee of the *National District Heating Association*, already referred to, recommends an allowance of 850 lb. per season of eight months per sq. ft. of effective radiation.

Heat Loss from Underground Steam Mains.* The *line losses* due to radiation from underground mains, properly protected and drained, range from 0.025 to 0.075 lb. of steam condensed per sq. ft. of underground pipe surface per hour, with 0.05 lb. as a fair average for low-pressure steam systems.

Actual tests on a system with wood-log insulation in use between 6 and 12 years, at an

* See also "Pipe Coverings," in the Chapter on "Pipe, Fittings, Valves, Coverings and Accessories."

average depth of 6 ft. under street pavement, and some concrete conduit with 1 inch asbestos lining in use 6 years, showed a condensation rate of 0.0511 lb. at 5 lb. gage and 0.0588 lb. at 25 lb. gage pressure.

Actual data from a typical distributing system using concrete conduit, wood log and tunnel with 85 per cent magnesia insulation ran as follows: The lines were run at an average depth of 6 ft. and had been in use 9 years, supplying steam summer and winter. See also Fig. 23.

TABLE 3

CONDENSATION TESTS

Month	Condensation per Square Foot per Hour	
June	0.014 lb. min.	0.024 lb. average 1st Section 0.047 lb. average 2nd Section
March	0.068 lb. max.	0.047 lb. average 1st Section 0.052 lb. average 2nd Section

Steam is often kept in the mains all summer not only to supply industrial needs, but in many cases to keep the lines dry, prevent corrosion of the pipe, and preserve the insulation.

The *presence of water* in the conduit or tunnel around the pipe will *increase the line losses enormously,* and cause the *rapid deterioration* of the entire underground system.

Construction and Installation of the Main Piping System. The *material* for the pipe lines may be wrought iron, mild steel, cast iron or even wooden stave pipe for returning the condensation to the central plant. The use of full weight wrought iron pipe is generally recommended, but if careful attention is paid to internal and external drainage, so that the *insulation is kept dry at all times,* the material for the pipe line seems to be of secondary importance.

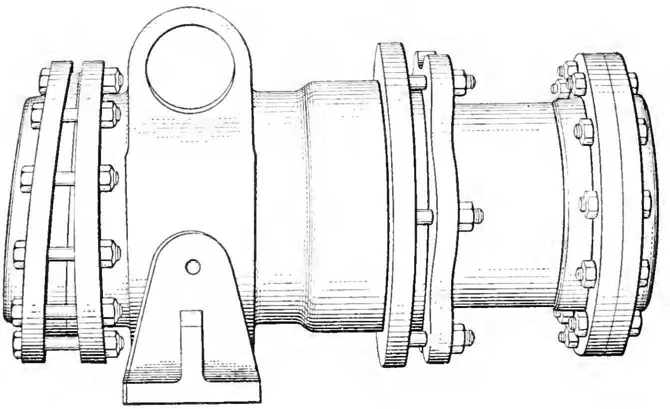


FIG. 7. SINGLE EXPANSION JOINT FOR UNDERGROUND WORK.

The *grading* and *draining* of the pipe line and the *underdraining* of the conduit is of prime importance. All unavoidable pockets or low points must be drained by steam traps, which are suitably located for frequent inspection. In city systems there should be a trap every two blocks.

In order to make the tile underdrainage effective, crushed stone should be placed around the conduit (Figs. 9, 11, 14 and 15) or at the base of the tunnel, extending down to the underdrainage, and the tile should be connected to the sewer at all low points. Unless suitable under-

drainage is provided the system may have to be replaced in from 7 to 10 years or less. Properly drained systems have been kept in use, and in good condition, for from 25 to 30 years.

Provision for *expansion and contraction* must be carefully considered, and at intervals of not more than 300 ft. the line should be *anchored* by means of suitable anchors, and between these points double slip-anchored expansion joints should be installed. See also "Expansion of Main Piping" under "Forced Hot Water Heating," in this chapter.

In placing these fittings great precautions should be taken so that only direct thrusts will act upon the expansion joints when the line is complete and the piping expands.

The method of *connecting pipe and fittings* must permit of extensions, removals and repairs, and hence flange joints should be used at all special fittings, while either flanged or screwed joints may be used on the runs.

The use of welded joints in connecting the long runs of underground piping is often found convenient, and insures a permanent and desirable type of joint. A committee of the *National*

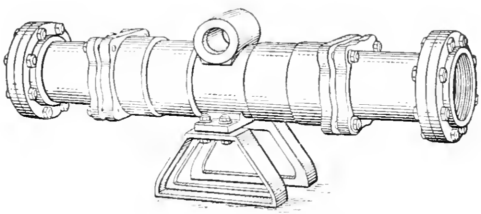


Fig. 8. Double Expansion Joint, Showing Method of Anchoring.

District Heating Association has submitted the following information (see the "Practical Engineer," Oct. 1, 1915) concerning this practice:

"Four-inch and smaller is usually welded on the bank; larger sizes are suspended above the trench and welded, or are laid on timbers in the bottom and welded in place, the welding being done on top and pipe revolved as it is welded.

"Costs will vary, but in common practice welding can be done at a cost 25 to 40 per cent less than screw couplings. For welding on the bank costs are about as given in the table. Welding in the trench will cost about twice as much.

TABLE 4

COST OF WELDING PIPE ON BANK

Size Pipe	Labor at 30c per Hour	Oxygen at 2c per Cu. Ft.	Acetylene at 2c per Cu. Ft.	Filling Wire at 12c per Lb.	Total Cost
4 in.	15″ $0.075	8 ft. $0.16	7 ft. $0.14	0.5 lb. $0.06	$0.435
6 in.	20″ .10	10 ft. .20	9 ft. .18	0.75 lb. .09	0.57
8 in.	35″ .175	20 ft. .40	18 ft. .36	1.0 lb. .12	1.055
12 in.	50″ .25	30 ft. .60	27 ft. .54	1.5 lb. .18	1.57
16 in.	90″ .45	40 ft. .80	36 ft. .72	2.0 lb. .24	2.21

"For welding 6-in. and 4-in. connections to 8-in. main, 90 deg.; cost of cutting $0.19, of welding $1.27, total $1.46.

"For 4-in. connections on 8-in. main, 90 deg., total $1.095.

"For 4-in. connection on 8-in. main, 60 deg., total $1.355."

Underground Conduit and Tunnel Construction. In central station heating systems it is generally necessary to run the distributing mains underground, and hence some sort of an en-

closing conduit or tunnel must be provided to protect the pipe and covering from the earth, and the ground water which is usually present in all soils to a greater or less degree.

A great variety of conduit constructions have been devised, and a number of patented commercial types are on the market. In many cases an attempt has been made to combine protection and insulation features in one material which could thus serve as conduit and covering at the same time.

FIG. 9. SEGMENTAL WOOD-LOG CONDUIT AND BASE.

The *segmental or wood log conduits* are notable examples of this sort, and while they are easily installed, and give a very satisfactory insulating efficiency when dry, they are liable to rapid deterioration if they are subjected to alternate wet and dry conditions. By reference to Figs. 9 and 10 it will be seen that 4″ thick radial staves of kiln dry white pine, carrying a tongue and groove on opposite sides, are bound together with $\frac{3}{16}$″ galvanized steel wire. This wire is wound on spirally under sufficient tension to embed it in the wood, leaving a practically smooth outer surface. The outside of the casing is finally coated with ¼″ of asphaltum pitch and rolled in sawdust. The inside of the casing is lined with 4A charcoal tin plate, and the ends of each section have four-inch mortise and tenons which are thoroughly creosoted and oiled. This form of conduit is approved by the *American District Steam Co.* and is known as their "Standard Construction." Dimension data are given in Table 5, which applies to both 4″ and 2″ casing.

TABLE 5

DIMENSIONS OF 4-INCH AND 2-INCH WOOD CASING

Size of Iron Pipe, Inches	Inside Diameter of Casing, Inches	Outside Diameter of Casing, 4-inch Shell, Inches	Outside Diameter of Casing, 2-inch Shell, Inches
1	3	11	7
2	4	12	8
3	5	13	9
4	6	14	10
5	7	15	11
6	8½	16½	12½
7	9½	17½	....
8	11	19	....
9	12	20	....
10	13	21	....
12	15	23	....
14	17½	26	....
16 OD	18½	28	....
18 OD	21	31	....
20	22¾	32¾	....

NOTE.—The iron pipe is to be made concentric with the wood casing by means of approved guides and rollers, spaced not to exceed 8 feet. The annular space, so far as possible, shall be made dead-air space by collars placed at ends of all anchorage fittings.

The great importance which this company attaches to suitable underdrainage will be at once apparent by reference to Fig. 9, showing double drain lines laid in crushed stone filler. The drain tile used ranges from 4″ to 6″ in size.

FIG 10. WOOD-LOG CONSTRUCTION.

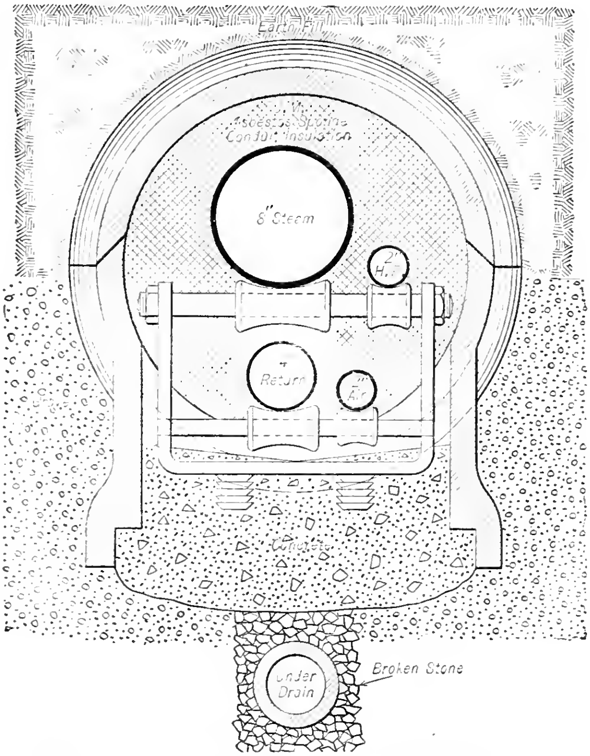


FIG. 11. JOHNS-MANVILLE CONDUIT AND BASE DRAIN.

The use of *vitrified split tile for conduits* has resulted in the development of a patented product, shown in section in Figs. 11 and 12. In the *Johns-Manville* conduit special anchor tees are used to support the piping on 12-,15- or 18-foot centers depending on size of pipe contained in the conduit. Thus, for 6″ or larger space 18 ft., for 2″ to 6″ pipe space 15 ft., and for 2″ or

smaller space 12 ft. on centers. The piping in this conduit is insulated by packing the air space between pipe and shell of conduit with some loose filling material, such as "Asbestos-Sponge Conduit Filling." The use of standard pipe covering is probably more satisfactory, as there is always

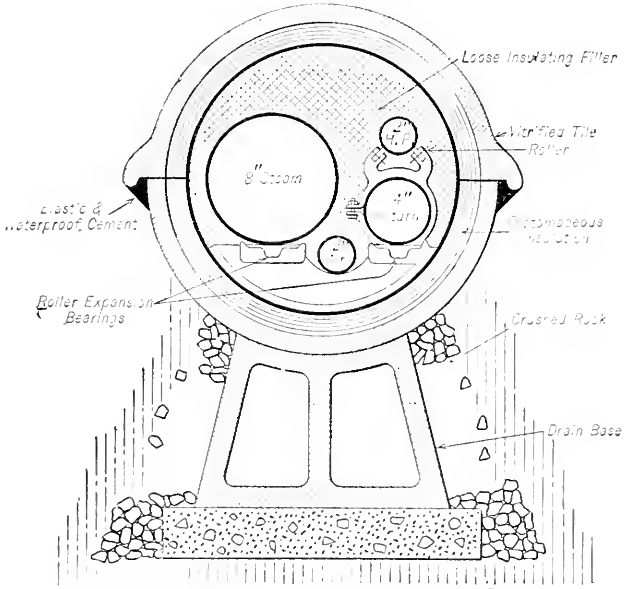


FIG. 12. "RIC-WIL" CONDUIT

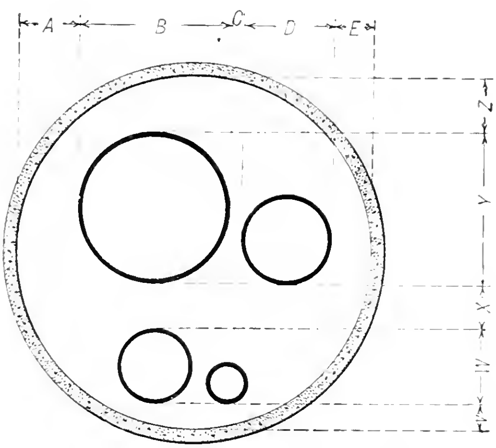


FIG. 13. SPACING FOR PIPES IN CONDUIT

a possibility that the loose filling may finally compress and settle down enough to uncover the top of the heating mains.

The size of conduit is determined by the *H. W. Johns-Manville Co.* as indicated below:

When four pipes are to be enclosed in the conduit it is believed that the formula given

below will be a satisfactory guide, in which it becomes necessary to consider the distance up and down as well as the distance across, and the greater distance only should be the guiding factor.

For illustration, assume the four pipes to be enclosed are 8″ and 4″ at the top, and 2½″ and 2″ at the bottom.

$A = 0.4B$.
B = external diameter of largest pipe on either level.
C = constant, 1 inch.
D = external diameter of other pipe on same level.
$E = 0.4D$.
$V = 0.5W$.
W = external diameter of largest pipe in lower row.
X = constant, 3½ inches.
Y = external diameter of largest pipe in upper row.
$Z = 0.5Y$.
A, E and Z must not be less than 3 inches.
V must not be less than 1½ inches.

First, the distance across the top (Fig. 13) equals $A + B + C + D + E$ = 3½″ + 8½″ + 1″ + 4½″ + 3″ = 20½″, using convenient approximate fractions.

The distance up and down = $V + W + X + Y + Z$ = 1½″ + 3″ + 3½″ + 8½″ + 4½″ = 21″.

The latter, being in this case the greater distance, would alone be considered in selecting the size of the conduit, and 22″ being the next size would be the one selected.

It is recommended that where more than two pipes are to be enclosed, the size be exactly determined by scale drawings, in the making of which the formula given will be of assistance.

Where more than four pipes are to be enclosed the size can only be determined by means of full size or scale drawings.

TABLE 6

J-M SECTIONAL CONDUIT

Inside Diameter of Conduit, in Inches	Conduit per Ft., in 3-Foot Lengths	Diameter (Internal) of Base of Support'g Tee, in Inches	Distance Center of Conduit to Center of 5-In. Underdrain, in Inches	Weight of Conduit per Feet, in Pounds	Approximate Weight of 2-Roll Frame, Pounds	Gross Interior Area of Conduit, in Square Inches	Thickness of Shell of Conduit, in Inches
6	$0.36	6	..	...	5	28.274	5/8
8	0.60	8	15	22	10	50.265	3/4
10	0.90	10	17	32	12	78.540	7/8
12	1.20	12	18	46	15	113.097	1
15	1.62	15	20	65	20	176.715	1 1/8
18	2.04	15	22	85	25	254.469	1 1/4
20	2.70	18	24	100	30	314.160	1 3/8
22	3.30	20	25	120	33	380.133	1 5/8
24	3.90	20	27	136	35	452.390	1 5/8
27	7.14	22	29	230	40	572.555	2 1/4
30	9.25	24	31	290	45	706.860	2 1/2

NOTES.—Supporting tee, each, 3 feet long.
Observation tee, each, 3 feet long, union, each, 3 feet long except 6-inch, 8-inch and 10-inch, which are 2 feet long.
Asbesto-Sponge Conduit Filling (approx. 7 lb. per cu. ft.) in 30 lb. bags; $320.00 list per ton.
27-inch and 30-inch are made double thick.
The 4-roll frames have rolls approximately 20 per cent. heavier than the 2-roll frames.

To determine the amount of insulating material required, consider each run of conduit separately. From the internal area of the conduit as given in the list, deduct the sum of the external areas of all of the pipes. Reduce the difference to square feet and multiply by the

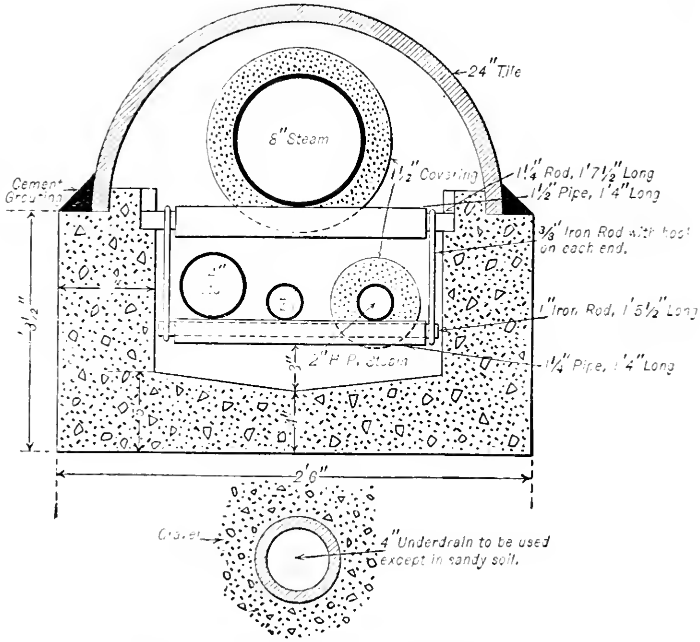


FIG. 14. TILE AND CONCRETE CONDUIT.

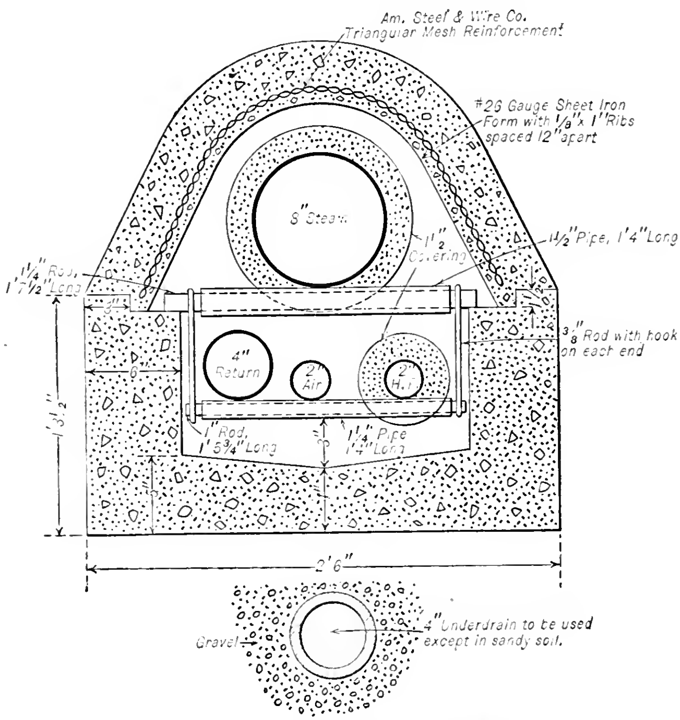


FIG. 15. SPECIAL CONCRETE CONDUIT.

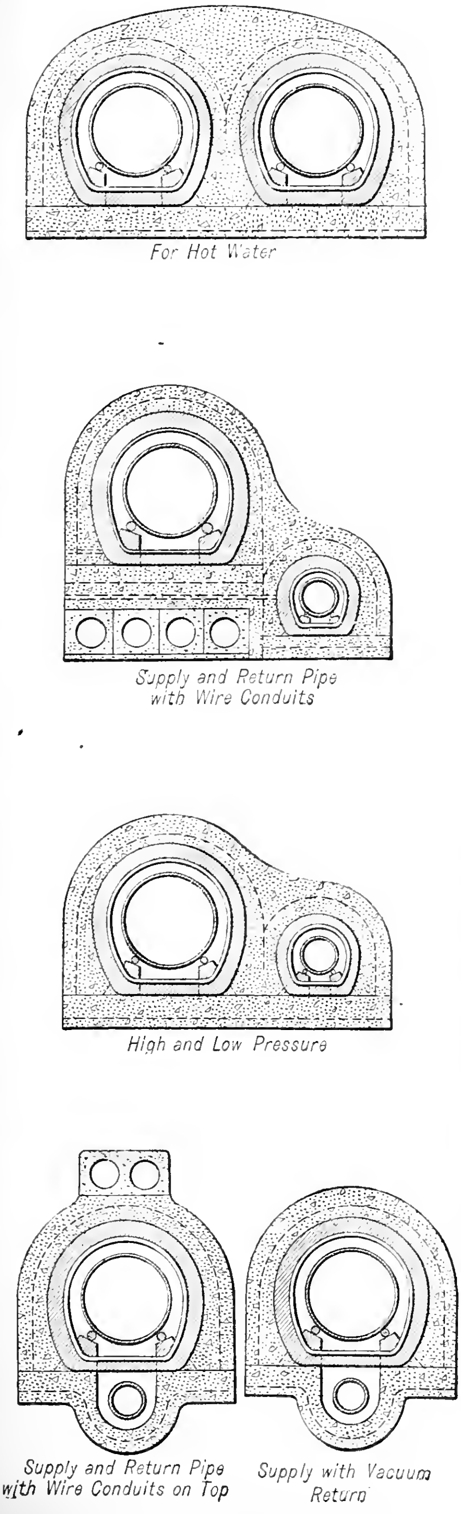


FIG. 16. SHAPES OF CRESCENT CONDUITS.

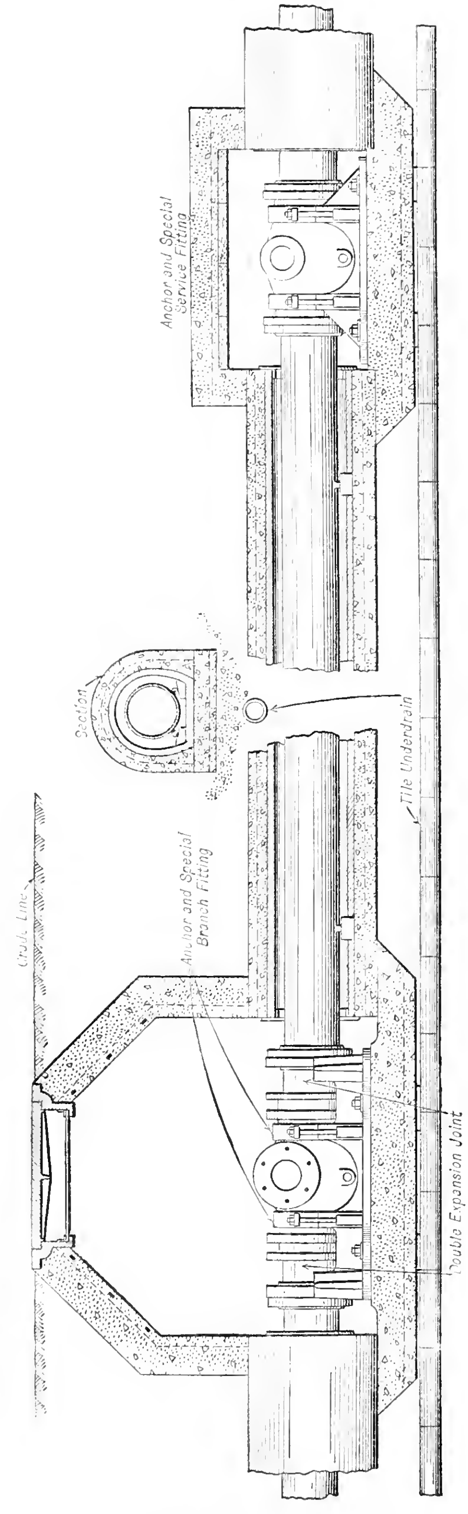


FIG. 17. CRESCENT SINGLE-PIPE UNDERGROUND CONDUIT.

length of the run in feet, and the result will be the cubic feet of space to be filled. Multiply this by the weight of the insulating material per cubic foot, which in the case of the "Asbesto-Sponge Insulating Material" is approximately seven pounds, and the result will be the pounds of insulating material required for the run.

The "Ric-Wil" Tile Conduit (Fig. 12) made by the *Richards-Wilson Co.* is somewhat similar to the conduit already described, except that it has an insulating layer of diatomaceous earth moulded on the inner face of the tile.

As will be seen in the sectional view, a patented *drain base* is used to support the split-tile, and the pipe supports are carried by cast-iron chairs or saddles resting on the interior lining of the conduit.

Modifications of these conduits with semi-removable covers are shown in Figs. 14 and 15; in the former a half-tile cover is used inverted above a substantial concrete base, which is suitably underdrained. In cases where this conduit may be subjected to heavy pressure the special arched form (Fig. 15) may be used, or the flat slab type (Fig. 18) substituted where conditions demand extra strength.

A *special moulded conduit construction* made by *W. H. Pearce and Co.*, is shown in cross-section in Fig. 16, and in application in Fig. 17. This is a patented conduit, and is made up in short lengths with a heavy tar or asphaltum-coated paper shell with tin lining surrounding *each pipe.* These shells form the core around which the concrete is moulded as shown.

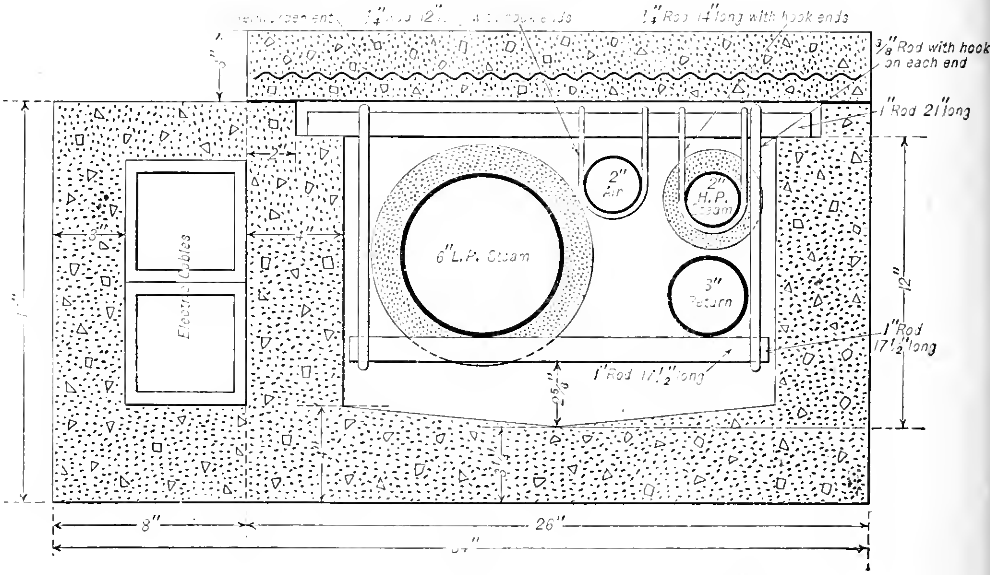


FIG. 18. RECTANGULAR CONDUIT FOR USE UNDER ROADWAYS.

The outer shell is of reinforced concrete—waterproofed on the outside. The conduit lining or insulation is made of materials possessing the highest insulating qualities. It is germ-proof and will not fall down or deterioate with age. It is lined on the inside with tin or similar material and is sufficiently large to allow a dead air space around the pipe in order to prevent pitting or outside deterioration.

The pipe is supported on dirt-proof ball bearings independent of the insulation, which allows free longitudinal movement without disturbing or injuring the insulation.

The expansion joints are set into a bored sole plate which supports the pipe flanges directly underneath, permitting of a longitudinal adjustment and allowing the body to be

tipped or adjusted to the angle of an outgoing branch. The enlarged body is a water separator from which condensation may be drained from the lowermost part. The sleeves operate loosely into the body, which with the ball bearings prevent sticking, strains on the pipe and consequent leaks. The anchors are of similar design, both permitting branch outlets to be taken from the sides.

The use of *rectangular conduits* (Fig. 18) with removable top or cover permits of placing and testing pipe before conduit is entirely finished. Sometimes it is possible to build the conduit complete in fairly long lengths if of suitable size, or in short lengths if small, and in this case the *box type* of conduit (Fig. 19), with permanent top, may be constructed with the pipe supports

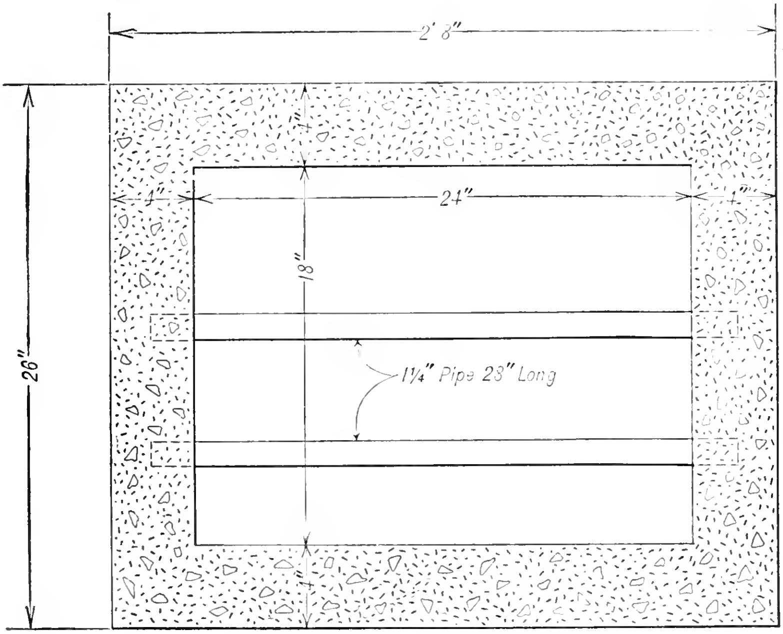


FIG. 19. BOX-TYPE CONCRETE CONDUIT.

and rollers all in place before any piping is installed. This applies to connections between buildings or short branches from large tunnels to buildings.

The use of *large conduits or tunnels* becomes necessary when a great many lines are to be run underground, or in case it is desired to provide for a variety of service lines in the same conduit or tunnel. Under the latter conditions it is most important that the tunnel section should be amply large to permit of workmen passing through the tunnel in order to make inspections and repairs. These tunnels may be rectangular or oval in cross-section, and are usually provided with structural steel framing on suitable centers for carrying the pipe supports, or else standard pipe brackets are carried on the side walls. In any event a clear space 6′–0″ high by 2′–0″ wide should be left as a passageway.

(3) The Steam Generating Plant

Capacity Required. The steam generating plant must be capable of supplying all the steam required by the radiation in coldest weather plus all losses due to transmission, feed-water heating, and boiler feeding. If a combination plant is contemplated, making use of exhaust steam,

a satisfactory working pressure of 100 lb. or more must usually be maintained, and suitable deduction made for the heat equivalent of the work done by the steam in passing through the engines. (See the Chapter on " Exhaust Steam Heating.")

Boiler capacity may then be computed as follows:

(1) Steam required for radiation installed in buildings heated to 70° is based on the condensing power of 1 sq. ft. of standard, three-column, cast-iron, direct radiation, 38″ high transmitting 250 B.t.u. per hour, or 0.25 lb. of steam per hour. Correction must be made for radiation of different type than that stated above or when operating under unusual temperature conditions, expressing same in terms of the standard, as given above.

(2) The line loss with properly proportioned and insulated underground mains will range from 5 to 10 per cent of the total steam transmitted at *full capacity*. This loss, of course, varies with the amount of underground piping and the percentage may be very large when the underground mains are transmitting only a small part of their total capacity. For example, with all steam practically shut off at each building served, but with steam in the tunnel or conduit mains, this condensation loss might exceed the amount of steam delivered to the buildings in mild weather. It may be taken at 0.05 lb. of steam per sq. ft. of main for good conditions with a gage pressure of 5 lb., as stated under " Heat Loss from Underground Steam Mains."

(3) Feed-water heating, if no condensation is returned from the buildings, may require 15 per cent of the steam generated, including station losses.

(4) The boiler capacity as determined by the power requirements must be based on the water-rate of the engines, and the load curve of the plant, but since at least 85 per cent of the heat in the steam used for power is available as exhaust for heating, we have only to see that the total heating requirements do not exceed the exhaust available at the time of maximum load in selecting the total boiler capacity. If the total heating requirements (Items 1, 2 and 3 above) do exceed the steam required for this maximum power load, then the heating requirements must determine the boiler capacity, plus one or more reserve units. (See the Chapter on " Exhaust Steam Heating.")

TYPICAL CENTRAL STEAM-HEATING SYSTEM

The following description and discussion of operating results of a central heating system, using exhaust and live steam, are taken from an article in " Domestic Engineering," August, 1908, by *James A. White*.

General Features. This is a small commercial district heating system of about 1000-boiler-horsepower capacity, of the single main type with condensation discharged to sewer.

" West Chester, Penn., a city of about 10,000 population, has a central station heating system which supplies heat to over 200 customers in business blocks, residences, public buildings, and churches.

" In the year 1902 it was decided to increase the scope and earning power of the plant by the addition of an underground system of pipes for the distribution of the exhaust steam from their engines for heating purposes, which, up to that time, had been allowed to go to waste.

" The location of the power house and the heating mains are clearly shown by Fig. 20. The power house is built of brick with concrete floors and wooden roof. A brick wall extending to the roof separates the engine and boiler rooms.

" **Boiler Room.** In the boiler room there are located three 250-horsepower and one 200-horsepower Stirling boilers, having a submerged heating surface of 9,500 square feet. All the boilers are equipped with Murphy furnaces and all but one of the 250-horsepower boilers have the Dutch oven type of furnace.

" **Coal Handling and Storage.** Coal is handled entirely by mechanical means, being emptied from the cars into a hopper below the tracks, and thence conveyed to the storage bins or conveyed to the bunkers above the boilers, each bunker having a capacity of 25 tons. The conveying machinery, built by the *Link Belt Engineering Company*, requires for its operation three

motors, having a total rating of 15 hp. From the bunkers the coal is carried through chutes by gravity to the magazines of the stokers. The ashes are wheeled away by hand. There is a storage room in the power house for 1000 tons of coal. The main coal storage space allows for a depth of 18 feet, which is dangerous for coal containing an excess of pyrites on account of

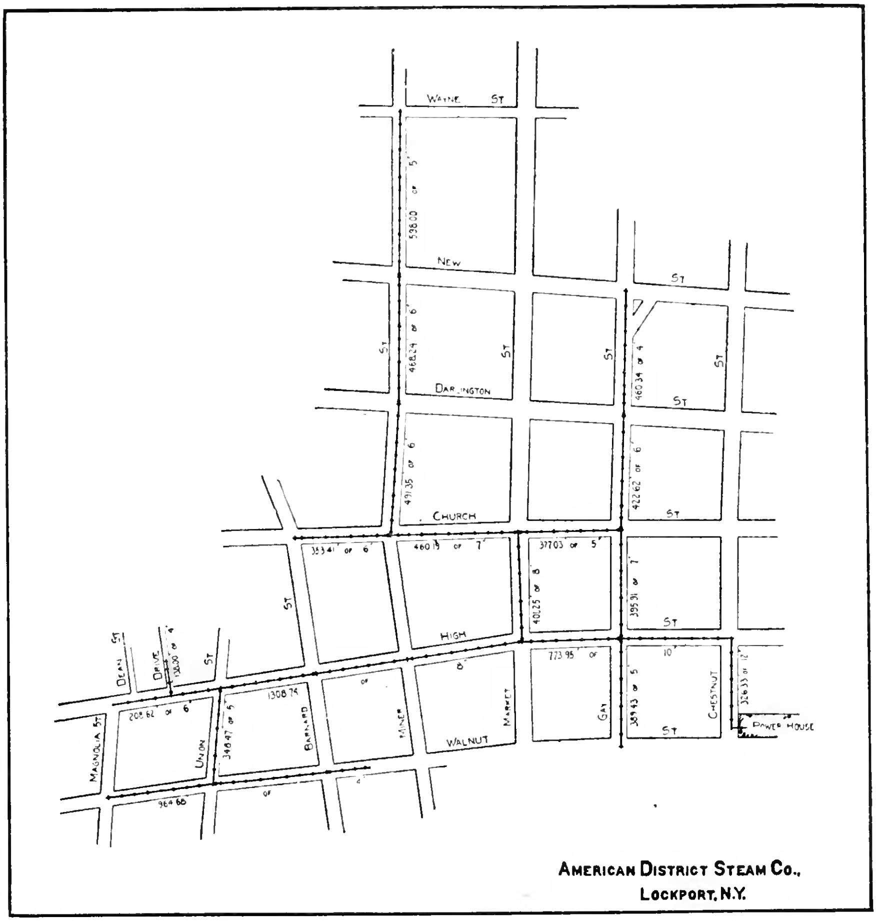


Fig. 20. Location of Power Plant and Heating Mains of West Chester Central Station Heating Plant.

spontaneous combustion. Ventilating pipes have therefore been placed in the bins, about 8 ft. center to center. These are made of 2 in. wrought-iron pipe with holes drilled about three feet apart protected by cast-iron hoods held in place by set screws. The tops of the ventilating pipes are arranged for attaching fire hose for flooding the coal pile in case the coal becomes overheated or takes fire.

"**Stack.** A brick stack 5½ feet in diameter by 150 feet high furnishes natural draft. The draft is regulated by a Patterson automatic draft regulator which is attached to the dampers, a change in the boiler pressure causing the regulator immediately to open or close the dampers.

"**Water Supply.** Water from the city mains is passed through a Thompson water meter and feed-water heaters before being admitted to the boilers. Two feed-water heaters are used, one 1000-hp. Cochrane open type, and one 600-hp. Berryman close type heater. The boiler feed pump is a Worthington, 7½ x 5 x 6-in. duplex, with the piping to the boilers so arranged that the fire pump, a 9 x 5¼ x 10-in. duplex *Barr Pumping Engine Company* pump, may be used in case the regular feed pump is out of commission.

"**The Engine Room.** The engine-room equipment consists of one 19 x 16-in. Ames engine, running at 225 r.p.m. direct connected to one 180-kw., 250-volt, D. C., three-wire machine. One 22 x 27-in. Ball Corliss engine running at 150 revolutions, direct connected to one 250-kw., 250-volt, three-wire generator. One 16-in. and 25 x 16-in. Ames compound engine running at 215 revolutions, belted to two 45-kw., 125-volt, three-wire system, *General Electric Company* generators, one 9B Brush arc, multi-circuit machine, and one 100-kw., 2-phase, Form P, 2300-volt, 60-cycle, General Electric alternating current generator. One 16 x 14-in. Ames engine, running at 230 revolutions, belted to two 45-kw., 125-volt, General Electric direct-current generators, and one old 150-hp. straight-line engine, belted to two 60-kw., Edison bi-polar machines, this latter engine being used as a reserve.

"The electrical output of the station is recorded by Thompson recording watt-meters, and the voltage is held constant by a General Electric T. E. voltage regulator.

"**Steam Piping.** The steam piping is so arranged that any boiler may be shut off from the engine side of the dividing wall and all valves in the piping system, except four in the exhaust pipe, may be repacked or removed and repaired without shutting down either the electrical or the steam heating plant. The exhaust piping is so arranged that all engines, or only two engines, can discharge their exhaust into the heating system. This is a very valuable feature, as it enables the station to be operated with back pressure on only those engines needed to supply the heating load at times of light demand, during the early and late parts of the heating season, the rest of the engines exhausting to the atmosphere without back pressure.

"The arrangement of the station piping is such that live steam may be fed to the heating mains at times when the power load is not heavy enough to supply sufficient exhaust steam, through one 2 x 4-in. Kiley and one 2½ x 5-in. Foster reducing valve, which delivers the steam to a booster connection. Steam may also be fed through an auxiliary 4 x 8-in. Foster reducing valve. There is a 12-in. oil separator placed between the station exhaust piping and the heating system to prevent the lubricating oil and water in the exhaust steam from entering the heating mains. An adjustable *American District Steam Company's* back pressure valve in the exhaust line holds the pressure required to supply steam to the ends of the heating mains. The maximum pressure required for the most severe weather is only 5 pounds at the station and this gives 1½ pounds at the extreme points of the system. All station piping is covered with 1½-in. magnesia covering, and all valves in the high pressure piping are heavy brass-seated angle valves.

"**Underground Work.** The underground heating mains, fittings, expansion variators, condensation meters and reducing valves used in the buildings heated were furnished and installed by the *American District Steam Company*, of Lockport, N. Y. None but strictly wrought-iron line pipe of full weight is used, and this is insulated against heat losses in a very thorough manner. The pipe is first wrapped with asbestos paper, which is held in place by spirally wound copper wire, and then inserted in a wooden casing, the inside diameter of which is about 2 in. larger than the outside of the pipe. (This casing is shown in Fig. 9 and its construction has already been described.)

"**Provision for Expansion.** As the temperature range in the pipes is over 180° Fahr., the linear expansion is considerable—amounting to about 1.4 inches per 100 feet. To allow for this expansion a very ingenious device called a Variator (Fig. 21) is used. This consists of a cast-iron frame or casing securely anchored to a brick-work pier or box, holding the outer edge of an annular corrugated copper disc, the inner edge of which is fastened to the free end of the pipe. These devices are placed about 100 feet apart with an anchorage fitting half-way between.

TABLE 7

TABLE SHOWING RESULTS OF OPERATING WEST CHESTER CENTRAL STATION HEATING PLANT FOR TWELVE MONTHS

Month	Lb. of Coal Burned	Cost of Coal	Total Lb. of Water Evaporated	Cost of Water	Lb. of Water Ev'porated per Lb. of Coal	Electric Output per Kw.-Hour	Lbs. of Water per Kw.-Hour	Lb. Steam Sold for Heat	Steam Not Sold for Heating		Income from Steam Sold for Heat	Cost of Boiler House Labor	Total Cost of Coal, Water and Boiler House Labor
									Total	Per Kw.-Hour			
July, '07	476,885	$538.47	3,646,433	$35.00	7.66	58,440	62.4	....	3,646,433	62.4	......	$147.00	$720.47
August, '07	516,325	541.91	3,046,065	29.30	5.9	63,994	47.6	....	3,046,065	47.6	......	147.00	718.21
September, '07	540,094	522.39	3,897,980	37.40	7.2	69,680	55.9	....	3,897,980	55.9	......	147.00	736.79
Total non-htg. mos.	1,533,304	$1,632.77	10,590,478	$101.70	6.91	192,114	55.1	....	10,590,478	55.1	....	$441.00	$2,175.47
October, '07	756,866	$786.06	5,451,120	$52.40	7.2	76,739	71.0	2,280,000	3,171,412	41.4	$914.22	$147.00	$985.46
November, '07	885,570	858.47	6,053,119	58.20	6.84	72,520	84.6	3,640,000	2,413,119	33.75	1,456.70	147.00	1,063.67
December, '07	1,157,940	1,243.55	7,322,804	70.30	6.34	82,365	89.0	5,340,000	1,992,804	24.24	2,132.73	147.00	1,460.85
January, '08	1,180,005	1,522.32	8,207,054	78.70	6.96	87,620	93.8	5,780,000	2,427,054	27.72	2,311.14	147.00	1,748.02
February, '08	1,233,560	1,597.00	8,694,744	83.50	7.04	79,430	109.0	6,540,000	2,154,744	27.17	2,617.79	147.00	1,827.50
March, '08	886,240	1,147.36	7,091,419	69.10	8.01	78,070	90.8	4,710,000	2,381,419	30.58	1,884.94	147.00	1,363.46
April, '08	668,460	819.79	5,310,542	51.00	7.95	66,023	80.4	2,950,000	2,360,542	35.8	1,180.32	147.00	1,017.79
*May, '07	509,167	624.29	5,302,341	50.90	10.4	65,248	81.2	1,258,000	4,044,341	62.0	503.66	147.00	822.19
*June, '07	487,053	534.78	3,820,806	36.70	7.86	58,164	65.7	635,000	3,185,806	54.8	254.43	147.00	718.48
Total htg. months	7,764,861	$9,133.62	57,254,241	$550.80	7.37	665,179	86.1	33,123,000	24,131,241	36.23	$13,255.94	$1,323.00	$11,007.42
Total 12 months	9,298,165	$10,766.39	67,844,719	$652.50	7.3	857,293	72.9	33,123,000	24,131,241	.....	$13,255.94	$1,764.00	$13,182.89

* Data for May and June, 1908, not available when record was made. May, $488.32; June, no sales. Total income, $12,987.17.

"As these variators require no attention after their installation, the brick boxes which surround them are filled with soft wood shavings for insulation and the pavement is replaced above. The numerous manholes which would be necessary in order to repack the ordinary slip joints are not required with this construction. Manholes are used, however, at street

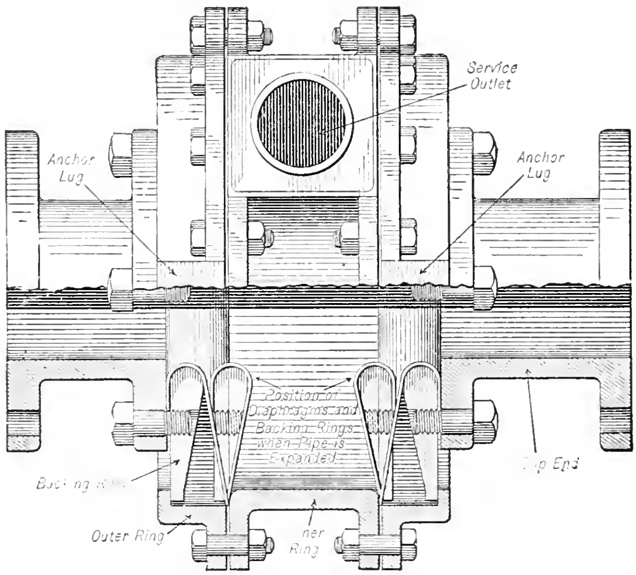


FIG. 21. INTERIOR VIEW, NEW TYPE OF DOUBLE "VARIATOR" EXPANSION JOINT.

crossings where valves are placed next to the crosses. The manholes have double covers with an air space between. The inner cover is arranged with packing, which prevents surface water from the street entering and coming in contact with the steam mains. Where the pipes are not of the same size at street intersections the crosses are made eccentric to secure proper drainage. The valves used are of the double disc, wedge-gate pattern, having a ground seat on the valve stem which relieves the stuffing box packing of all steam pressure when valve is full open or closed. The cut, Fig. 17, shows some of the details of this type of construction.

"**Services to Buildings.** All services to the buildings are taken from the top of the anchorage fittings and the stationary part of the variators at a point well above the top of the steam main. Taking the supply of steam from the mains in this manner insures the delivery of dry steam to the customers. The steam mains are carefully graded to low points where the water of condensation is trapped off from either inverted anchorage fittings or the bottom of the variators. In certain plants, using similar construction underground and having all drainage outlets equipped with meters for measuring the condensation taking place in the mains, the loss from this cause has been found to be less than 5 per cent of the steam delivered to the consumers during the heating season. A comparison of this figure with the losses in either gas or electrical transmission system will be found very favorable to the heating system.

"**The Electrical Load.** The electrical load attached to this system consists of 335 hp., in motors—68 motors of ½ hp. and larger being connected, the equivalent of 14,000 16-cp. incandescent lamps, 104 street arc lamps and 90 commercial arc lamps, 100 flatirons, besides a considerable number of fans, sewing-machines, dental engines, coffee-mills, etc. The comparatively small power load attached to this station is due to the fact that West Chester is more of a residence than a manufacturing town.

"**Steam Piping in Buildings.** There are about 8,900 feet of underground steam mains installed, from which steam is supplied to 212 customers, whose buildings contain about 100,000 square feet of radiation. About 70 per cent of the buildings heated are equipped with the atmospheric system, and the rules of the company require that all new customers have their buildings piped so this system may be used. Fig. 1 shows the method of piping for this system. (The operation of this system has already been explained.) Owing to the low pressure required in the customers' houses, due to the use of the atmospheric system, the heating company is able to give satisfactory service with only $1\frac{1}{2}$ pounds pressure at the end of their steam mains, to secure which but five pounds back pressure is carried on their engines in the coldest weather.

"**Operating Results.** A study of Table 7 of results of operation for a period of twelve months reveals some very interesting facts. It will be noted that during three months when no heating load was carried each kw.-hour produced required the evaporation of about 55 pounds of water, while during the heating months 86.10 pounds of water were evaporated per kw.-hour produced—this larger amount being due in part to the live steam required to supplement the exhaust from the engines, when the heating load was in excess of the electrical load, and in part due to the increased steam consumption by the engines due to the back pressure required to circulate the steam in the underground mains.

"It should be further noted, however, that during the heating months there was sold for heating purposes about 33,123,000 pounds of the 57,254,241 pounds of water evaporated, leaving 24,131,241 pounds of steam or water chargeable against the generation of electricity, and to care for any condensation or other line losses in the heating mains. During the same months there were 665,179 kw.-hours produced. Dividing the 24,131,241 pounds of water by 665,179 gives but 36.23 pounds of water evaporated chargeable to each kw.-hour produced during the heating months against 55 pounds chargeable in the non-heating months.

"To put it another way, 53 per cent more steam is chargeable to the manufacture of a kw.-hour during the three months when no back pressure is on the engines than during the nine heating months when the engines are carrying the additional load due to the back pressure required for circulating the steam for the heating system. During the month of December only 24.24 pounds of steam per kw.-hour is chargeable to the generation of electricity, due to the amount of steam sold for heating purposes. If the space heated had been extensive enough to require for heating purposes all of the steam manufactured, the electricity generated by the engines in reducing the steam from boiler pressure to that carried on the street mains could have been considered a by-product in so far as boiler room costs were concerned.

"A factor, tending to reduce the good showing of this company, is the nature of the electrical load, which consists chiefly of lighting. The daily electric load is very different from the heating curve which necessitates the supplying of much live steam to the heating system during parts of the 24 hours, while at other hours, even in the coldest months, exhaust steam has to be thrown away to the atmosphere. Nevertheless a comparison of the income received from the steam sold during the nine heating months with the cost of operation reveals the fact that the heating receipts were sufficient to more than pay for the coal and water as well as all other boiler-house labor for the entire year, during which time there were produced and sold from the same coal, water and boiler house labor 857,293 kilowatts of electricity.

"Results such as these are dependent in a large measure upon favorable location of the power house for the distribution of steam for heating purposes, and engineers cannot too carefully consider the location of proposed power houses in view of the advantages to be secured from selling the heat contained in the exhaust steam from their engines."

CENTRAL HOT-WATER HEATING SYSTEMS

Operating Conditions. Hot water may be readily circulated through suitable distributing mains from a central plant to buildings equipped with hot-water radiators and the necessary

service piping. Either a one- or two-pipe system of mains may be employed with suitable shunt circuits in the first case or with branch connections to separate flow and return mains as in the latter case. The two-pipe system is more common.

The circulation is usually maintained by a centrifugal pump or pumps located in the central plant, and connected to the return side of the system, in order that it may operate on cooler water.

The water may be heated by passing it through water-heating boilers, or tubular closed-water heaters, or "comminglers" of the injector type in which the steam used is injected into and condensed by the circulating water.

The temperature of the circulating water depends on the permissible pressure at the highest radiator, which for open-tank systems should not exceed two pounds above atmosphere, and for closed-tank systems should not exceed fifteen pounds gage pressure. On this basis a maximum of about 215° F. would be possible in the first case, and about 240° F. in the second case would be the upper limit of the outgoing water temperature. The lower or return temperature is usually about 30° less than the flow temperature in forced hot-water work.

In either system the water will be circulated, during a large part of the time, at much less than the maximum temperature, due to the fact that the average outside air temperatures are much above the minimum for which the radiation was proportioned.

Essential Features of a Hot-Water System. As in the case of steam systems there are three subdivisions, including: (1) The building equipment of hot-water radiators and piping with control valves; (2) The distributing mains and tunnels or conduits, with all necessary expansion joints, valves, chokes, and service connections to buildings; (3) the central heating plant equipment of boilers, heaters, circulating pumps and expansion tank.

(1) Building Equipment

Radiators and Piping. The size of radiators is determined as already indicated in the Chapters on "Heat Transmission of Building Construction and Direct Radiators," allowing a mean water temperature of 170° F. and a heat transmission of 150 to 170 B.t.u. for low-pressure systems and a mean water temperature of 215° F. and a heat transmission of 250 to 270 B.t.u. per sq. ft. per hour for high-pressure systems.

Since it is customary to allow a drop of 30° F. in the temperature of the water passing through the radiators, it will be apparent that each gallon of water gives up $30 \times 8\frac{1}{3} = 250$ B.t.u. between the inlet and outlet. If the heat loss from the buildings supplied by water from the central plant is known, and is represented by H, then the theoretical equivalent amount of water to be circulated per hour is $H/250$ gals. $= G$.

The *actual amount of hot water required* for forced hot-water systems depends on how carefully the flow is regulated. For hot-water systems, it was found that the use of differential control valves to regulate the circulating pressure, and temperature control to regulate the flow, resulted in marked economy. Many stations reported from 6 to 8 lb. of water circulated per hour per square foot of radiation, while with the two controls satisfactory heating was accomplished with from 3.5 to 4 pounds circulated per hour. Tests showed that with proper control more than 6 pounds circulated per hour was needless, and that water pumped above this amount is all out of proportion to the increased heat derived.

Uncontrolled circulation may run as high as 14 pounds per hour per square foot, with little benefit gained. Waste of heat on uncontrolled systems will run from 20 to 100 per cent of the amount actually necessary.

This amount of water must also supply the heat losses from mains, service lines and building piping, and hence must undergo a greater total drop in temperature than 30° F., which is to be allowed through the radiators. If the piping loss is in B.t.u. per hour H_m, then the total drop in temperature will be $(30 + t)$, where $t = H_m/(G \times 8.33)$.

For average weather conditions a much lower mean water temperature is desired as well as

a smaller drop in temperature through the radiators, in order to prevent overheating of the buildings.

This modification of the temperature of the circulating water is of great importance from the standpoint of economy of operation, since the average outside temperature during the usual heating season of 200 days ranges from 30° to 40° F. above 0° F.

The following table of outgoing water temperatures provides for open- and closed-tank systems and has been found satisfactory in practice. An allowance for wind velocity should also be made as indicated.

TABLE 8

SCHEDULE OF OUTGOING WATER TEMPERATURES, FORCED HOT-WATER SYSTEM

Table (A)		Table (B)	
With Open Expansion Tank		With Closed Expansion Tank	
Outside Temperature	Water Temperature	Outside Temperature	Water Temperature
Degrees F.	Degrees F.	Degrees F.	Degrees F
50	140	50	150
45	140	45	155
40	150	40	160
35	160	35	165
30	165	30	170
25	170	25	175
20	180	20	180
15	185	15	185
10	190	10	190
5	195	5	195
0	200	0	200
−5	205	−5	210
−10	210	−10	220
−15	215	−15	230
−20	215	−20	240

Add ½° F. for each mile wind velocity over 5 miles per hour.

By controlling the outgoing water temperatures as shown by the above tables, satisfactory temperature control may be secured throughout the system without the usual thermostatic system of regulation usually necessary in every building supplied by a central steam system.

An actual schedule of outgoing water temperatures is given in Table 9, which provides for higher temperatures for Sundays and holidays.

TABLE 9

TEMPERATURE SCHEDULE (IN DEGREES FAHR.) FOR HOT-WATER-HEATING STATION

(*Merchants' Heat and Light Company*, Indianapolis, Ind.)

Outside Temperature	Temperature of Flow	Outside Temperature	Temperature of Flow
−8	200	30	157
−5	196	40	145
0	190	50	131
+5	184	60	115
+10	178	65	105
20	167	70	95

Note.—On Sundays and holidays maintain a schedule 5 deg. Fahr. higher than the above. Also add one degree for each 2 miles of wind velocity.

(2) Distributing Systems

The systems of main piping in most common use are the one-pipe circuit, or belt system, and the two-pipe multiple system.

One-Pipe System. The one-pipe system is especially desirable where the buildings are arranged in circuit with the last one in the series near the central plant. With this system the service connections are shunts taken off the main at the top and returned into the side or bottom of the same main. Circulation is produced in the shunt by introducing a choke or valve in the main between the service connections, or, in case the main runs through the basement of the building, the flow and return connections may be kept apart by the length of the building to provide sufficient resistance in the main to cause flow in the shunt, and thus supply the building.

Two-Pipe System. The two-pipe system consists of a separate flow and return main, radiating from the plant as two parallel lines, usually in the same conduit or tunnel, and of the same diameter, the size of each main being changed whenever the branch or service connections are large enough to permit the use of the next commercial pipe size in the mains.

Comparison of One- and Two-Pipe Systems. A comparison of the two systems shows the initial costs to be about equal, although the one-pipe system is, of course, cheaper, and the economy of operation, including line losses is also about the same for both. The two-pipe system is more flexible and the service rendered more uniform. Service connections are smaller for this system, but station conditions, depreciation and maintenance are about the same for both. The two-pipe system is probably more commonly used, especially in extensive systems, and is much better adapted for serving contiguous buildings as in city blocks. Either system may be applied to undulating ground conditions, provided service connections are taken off at all high points to relieve the air which would otherwise be trapped in the mains at such points. If this is not feasible then special air valves must be used at these points. The system should not be applied under conditions where the maximum difference in head exceeds 100 ft. if cast-iron radiators are to be used.

Sizes of Distributing Mains, Based on Quantity of Water and Velocity. The size of the distributing mains in a forced hot-water system depends on the quantity of water to be circulated per hour or per minute, and the allowable friction pressure loss in the system, which in turn is a function of the velocity head.

The *quantity of water* to be circulated is determined as already indicated earlier in this chapter, and is based on heat loss from buildings supplied, but an additional allowance in temperature drop for each gallon circulated must be made for radiation losses from the mains during transmission to and from the radiators, which loss ranges from 5 to 10 per cent of the heat transmitted by the mains when operating at maximum capacity.

As soon as the quantity of water W in pounds per hour has been determined, the size of main for any assumed velocity V, in feet per second, is readily found. Thus the area of the main A in square feet is: $W/(V \times 3600 \times D)$, where D is density of water at mean temperature of flow, or approximately 60.5 lb.

The *assumed or allowable velocity of flow* in forced hot-water systems depends on the permissible friction pressure loss, which latter must be determined largely by considerations of economy of installation as balanced by economy of operation. The use of smaller pipe and higher velocities making for the former, and larger pipe and lower velocities making for the latter condition, except in the matter of radiation losses, which will be less with the smaller pipe.

For a complete discussion of this matter see *Arthur M. Greene's* "Heating and Ventilation," pages 153 to 170.

The values used in practice range from 5 to 10 ft. per second, the former being used for velocities in branch connections and outlying mains and the latter for trunk mains.

The velocities in one-pipe mains are made from 2 to 3 ft. per second higher than in two-pipe mains, and a velocity as high as 12 ft. per second may be used in the former. In this system

the cooled water from one building becomes in part the supply for the next, so that higher velocities are used. In applying the above velocities care should be taken to see that the amount of water to be circulated is the maximum that will be required by all future extensions to the system.

The following velocities are recommended by *C. L. Hubbard* for two-pipe work, and are conservative:

TABLE 10

MAIN VELOCITIES FOR FORCED HOT WATER

Diameter of Main	Velocity in Feet	
	Per Minute	Per Second
3″	250	4.
4″	300	5.
5″	350	6.
6″	400	6.7
7″	450	7.5
8″	500	8.3

Friction Pressure Loss in Forced Hot-Water Systems. The following table from *N. S. Thompson's* "Mechanical Equipment of Federal Buildings," gives the friction head in feet of water at 180° F. for 100 ft. of main, and is based on the formula $h = \frac{0.32V^{1.86}}{10D^{1.25}}$, in which V = ft. per sec., D = diam. in ft., h = head in ft. per 100′. $h \Big/ \frac{144}{60.5}$ or $h / 2.38$ = friction head or loss in lb. per sq. in. It is quite customary to add 10 per cent to the measured length of pipe to provide for the additional loss in fittings, valves, etc. Pipe bends or long radius fittings should be used wherever possible.

TABLE 11

FRICTION HEAD LOSS IN FEET PER 100 FEET FOR FORCED HOT-WATER MAINS

$$h = \frac{0.32V^{1.86}}{10D^{1.25}}$$

h = loss in feet per 100 feet.
D = pipe diameter in feet.
V = velocity in feet per second.

Pipe Size, Inches	Velocity in Feet per Second								
	2	3	4	5	6	7	8	9	10
2	1.09	2.32	4.06	6.02	8.46	11.25	14.43	17.96	21.88
3	0.658	1.40	2.45	3.62	5.08	6.76	8.70	10.80	13.20
4	.459	0.980	1.70	2.52	3.54	4.70	6.04	7.60	9.17
5	.349	.742	1.29	1.92	2.70	3.57	4.60	5.70	6.97
6	.276	.588	1.03	1.52	2.14	2.83	3.64	4.53	5.53
7	.228	.485	0.848	1.25	1.75	2.34	3.00	3.72	4.56
8	.193	.411	.717	1.06	1.50	1.98	2.54	3.17	3.86
9	.166	.354	.617	0.913	1.28	1.70	2.19	2.72	3.33
10	.146	.311	.541	.800	1.12	1.50	1.93	2.37	2.92
12	.116	.246	.431	.638	0.90	1.19	1.53	1.91	2.32

The friction pressure loss to be overcome by the pump will vary from 25 to 50 pounds per sq. in., depending on the size of the installation, and the static head on the pump should not exceed 45 pounds. When the pump is in operation the outflow pressure will rise to possibly 65 pounds and the return will drop to 25 pounds, giving a differential of 40 pounds required to overcome friction in the piping system.

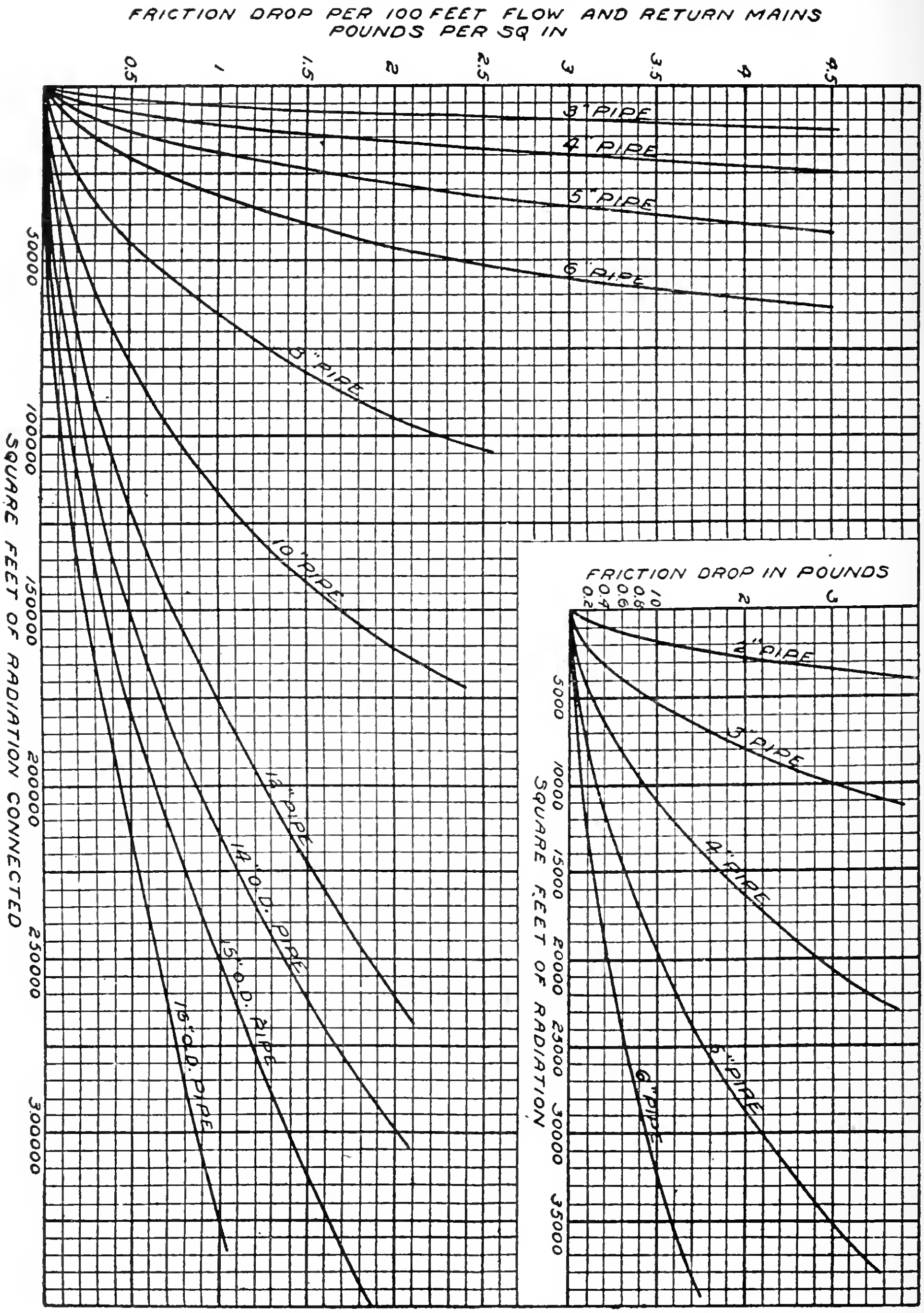


FIG. 22. FRICTION DROP IN FORCED HOT-WATER HEATING MAINS.

Example. The following example will illustrate the use of the above table: *Given*, 20,000 sq. ft. of radiation transmitting 170 B.t.u. per sq. ft. per hour at a distance of 100 ft. from the pump. Loss of head to be 1 ft., and water in radiation to cool 30° F.

Find. Size of flow and return main desired: $20,000 \times 170 + 5$ per cent for line losses $= 3,570,000$ B.t.u. per hour. Also $3,570,000 / (30 \times 60.5 \times 3600) = 0.546$ cu. ft. per sec. By trial of several pipe diameters it will be found that a 6-in. pipe, area 0.196, gives the only velocity corresponding to 0.5 ft. loss in head per 100 ft. This velocity $= 0.546 / 0.196 = 2.8$ ft. per sec. Referring to Table 11, gives by interpolation a 6-in. main with 0.5 ft. loss per 100 ft. or, 1 ft. loss per 200 ft. of run.

The curves shown in Fig. 22 have been approved by the *National District Heating Association* and have a general application to any forced hot-water problem. By referring to same it is

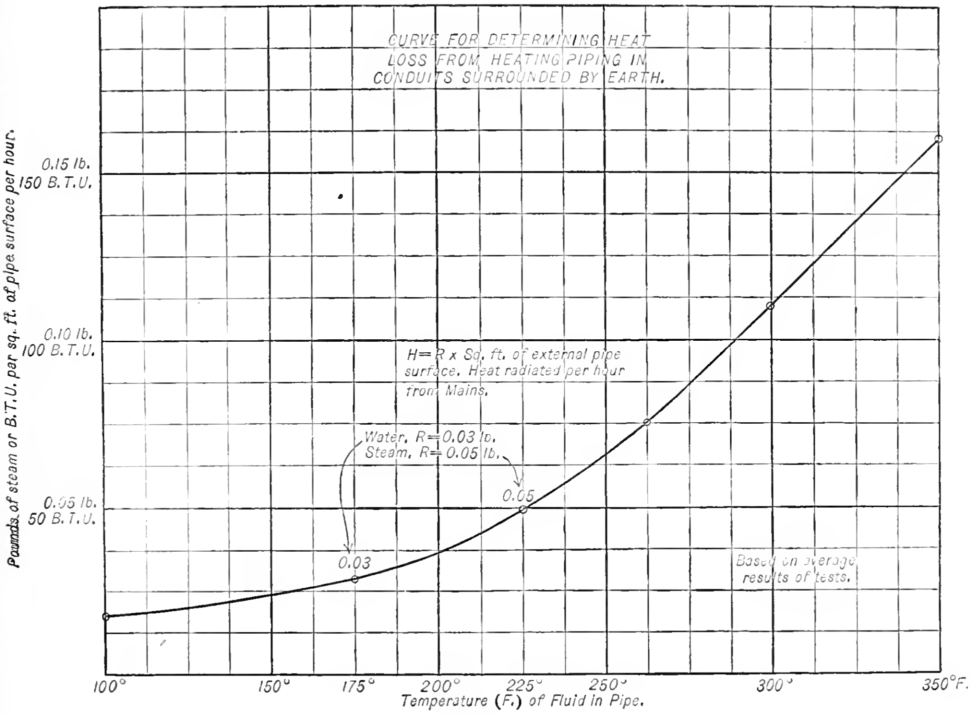


FIG. 23.

possible to select by inspection the proper size of pipe to supply a given quantity of radiation (allow for transmission loss) with any desired pressure drop in pounds per sq. in. per 100 ft. of main. The radiation is assumed to transmit 170 B.t.u. per hour with a 30° drop in temperature, water entering radiator at 180° F.

In two-pipe work allow for decrease in friction by loss of water at branch connections between the 100 ft. points.

Heat Loss from Mains in Underground Conduit. The allowance for heat loss from the underground mains depends as in the case of steam on the character and condition of the insulation and conduit used for enclosing the distributing piping as well as the actual amount of surface installed. For average conditions this loss may be determined by reference to Fig. 23, and may be expressed in pounds of steam condensed or in B.t.u. transmitted per hour.

Branch Connections or Secondary Distributing Systems. The branch or service connections to separate buildings depend upon whether the main system is of the one- or two-pipe type, and are from 1½ in. to 2½ in. in diameter. In *one-pipe work* these branches are taken off as shunts, of the same pipe size from end to end. "Y" fittings are used where the shunt leaves and returns to the main, in order to induce flow through same. These points of connection are kept as far apart as possible, and very often a valve or choke is inserted in the main between the two ends of the shunt to control the resistance.

It is considered good practice to carry the one-pipe main through the basement of the building if the use of the building will permit.

In the case of separate buildings, where the radiation will never be shut off, the entering shunt main may be divided into a number of circuits of radiators in series, and the combined sectional areas of the pipe connections made approximately 50 per cent greater than that of the main. The divided circuits reunite into a single main before leaving the building. The circulation through the radiators in low buildings may be forced by adopting the above method.

In *two-pipe work* there is always sufficient difference in pressure between the flow and return mains at any service connection to establish positive circulation, and the secondary system may be designed as for one-pipe work, although it is usually advisable to provide for equal length of water travel to all radiators to prevent short-circuiting.

For forced hot-water systems, no service pipe smaller than 1¼ in. should be used, and radiation can be connected satisfactorily on the following basis:

TABLE 12

RADIATION SUPPLIED BY HOT-WATER SERVICE PIPE

Service	Radiation, Sq. Ft.
1¼ in.	0 to 1,000
1½ in.	1,000 to 2,000
2 in.	2,000 to 4,500
2½ in.	4,500 to 6,000

The secondary systems for *high buildings* are usually of the overhead type, the service main being carried to the top of the building with distributing main in attic. Drop risers of the one-pipe type connect with the return service main in basement. The flow through the individual radiators is caused by gravity only, " Y " fittings being often used where radiator branches connect to risers. The flow connection to the radiators should be at the top and the return at the bottom, on the same or opposite ends.

Secondary systems in high buildings cause so great a static pressure on the lower radiators and pumps that their use is limited to the first ten or twelve floors. If upper floors are to be served with forced hot water, then separate pumps and heaters must be installed at the upper level, and steam connections made to these separate sets, each set taking care of from eight to ten floors.

Expansion of Main Piping. The main piping system must be designed to provide for contraction and expansion, which will vary with the temperatures carried in the mains. The coefficient of expansion for wrought iron is 0.00008″ per foot per degree, so that for a change of temperature from 32° to 212° F. the change in length per 100 feet of main is 1.44″.

This change in length becomes a serious factor when considered in connection with the high pressures in forced hot-water systems in long runs of piping, and must be definitely provided for by the use of suitable slip or corrugated expansion joints, or by the use of swing joints using 90° ells or spring offsets. (See Figs. 7, 8 and 21.) These joints or offsets should be placed from 350 to 500 feet apart, and the mains anchored midway between these points, which in cities may be placed at the crossing of intersecting streets.

Slip joints are easily installed, but require packing and attention. Corrugated or flexible joints are liable to rupture and offset, or swing joints require large lateral chambers, as the offset is usually made from 25 to 30 pipe diameters in length, and the bends made with long radius elbows.

In the use of offsets it is customary to place them under an initial strain when cold equal to one-half the total expansion which will occur when hot. This is done by cutting the last length of pipe in the main run short by this amount and then springing the offset cold until connection is made.

A maximum movement of 5 inches may be allowed for each expansion joint, which determines the maximum spacing of same. These joints should be as simple as possible and if of the slip joint type should have cast-iron body, brass sleeve, and be fitted with a metallic packing which is easily renewed.

Underground Conduit and Tunnel Construction. See the discussion of this subject as given earlier in this Chapter under "Central Steam-Heating Systems."

(3) The Central Hot-Water Heating Plant and Its Equipment

The equipment of the central hot-water plant must provide heaters for reheating the circulating water, as well as pumps for forcing the water through the system against the friction head, and this equipment is practically the same for either the one- or two-pipe system, but varies as to details, depending on whether the open or closed type of expansion tank is used.

Methods of Heating the Circulating Water. A central plant designed for hot-water heating only is usually provided with high-pressure boilers to furnish steam for driving the circulating pumps, the exhaust being used in an *exhaust heater* and condensed, and *water-heating boilers* are installed in series with the exhaust heater to supply additional heat when needed. As in all forced hot-water systems, the pump suction is connected to the cooler return water and the discharge delivered through the heaters or boilers and then into the flow main.

A central hot-water plant, designed to make use of exhaust steam as a by-product, must have the exhaust connections from all apparatus made into an exhaust heater, and must also have a *live-steam heater,* usually in series with the exhaust heater, of sufficient capacity to supply all the heat needed by the system with the exhaust heater out of commission.

In many plants additional provision is made for heating the circulating water, by the use of comminglers, economizers and special boilers.

The *Commingler* is practically a low-pressure injector in which the exhaust steam, freed of oil, is mixed directly, as in an open feed-water heater, with the circulating water.

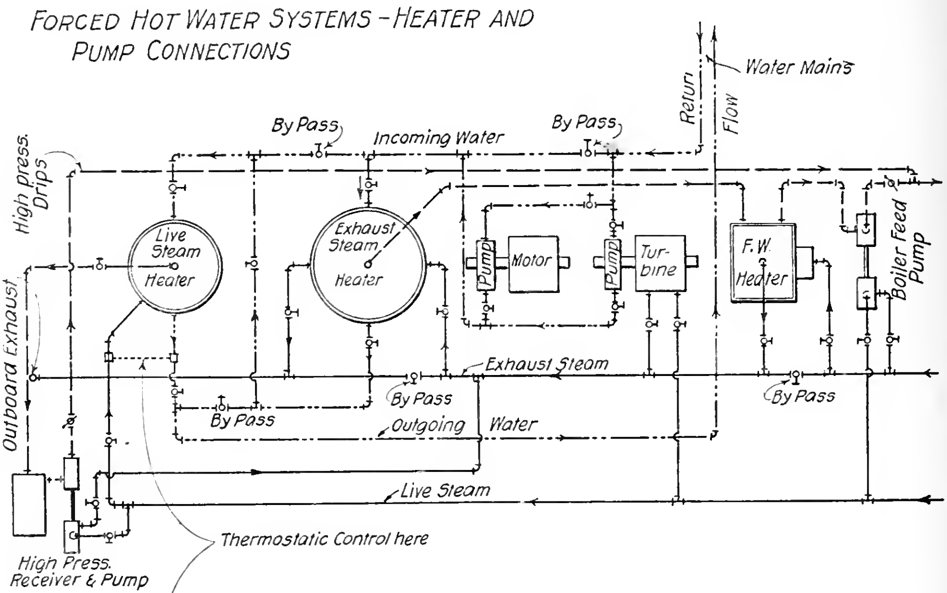
The *Economizer,* placed in the smoke flue from the boilers, has a limited use in forced hot-water systems. Unless kept clean its efficiency falls off rapidly with length of service, and there is constant need of repairs to apparatus of this sort, which accounts for its restricted application.

Heaters for Live and Exhaust Steam. The heaters in common use are made up with cylindrical steel shells with bolted heads. The *heating surface* consists of charcoal iron tubes $1\frac{1}{2}''$ or $1''$ in diameter or corrugated brass or copper tubes may be used. These tubes are expanded into partitions or tube sheets of steel plate. The tubes are usually staggered to prevent weakening the tube sheet as much as possible.

The heaters are almost invariably placed on end, in order to provide for uniformity in expansion of the tubes in case of partial flooding of the heater with condensation, as well as to economize floor space. They should, of course, be placed as close to the pump discharge as possible, and properly by-passed, so that any heater or combination may be cut out of service. See Figs. 24 and 25 for a diagrammatic layout.

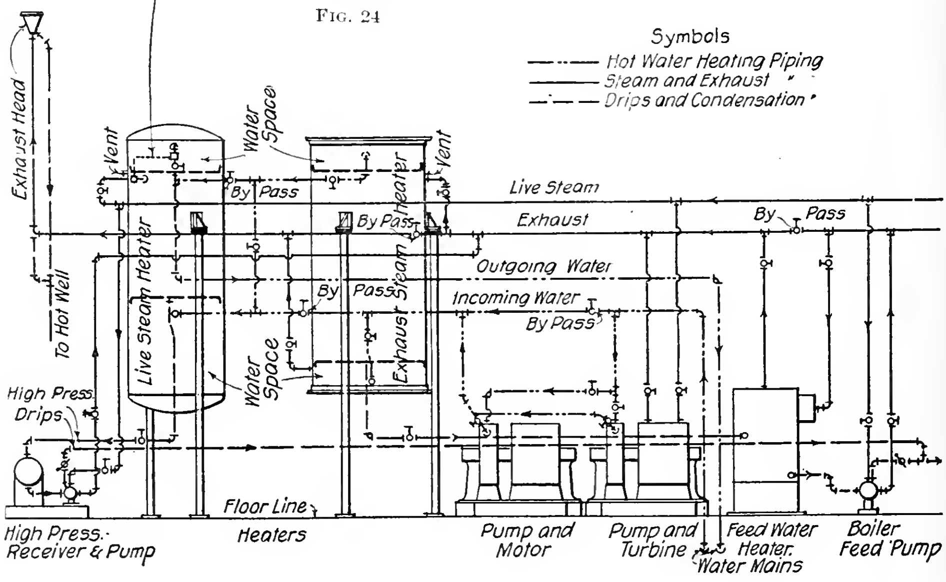
Steam enters the heaters at the top and surrounds the tubes, the water flowing upward, so that the hottest water comes in contact with the hottest steam, giving maximum efficiency for heat transmission.

Determination of Size and Heating Surface for Heaters. The *transmission rate,* on which



Plan of Heater and Pump Connections

FIG. 24



Elevation of Heater and Pump Connections

FIG. 25

the tube surface of exhaust heaters is determined is commonly based on 6,000 B.t.u. per sq. ft. per hour for steel tubes, and 7,000 B.t.u. for copper or brass tubes. A more exact method of proportioning heater surface is to use the proper transmission coefficient of the material as described below.

The *transmission coefficient* for the heating surface in these heaters ranges, for steel or wrought-iron tubes, from 180 to 270 B.t.u. per sq. ft. per degree difference per hour, depending on whether the water is on the outside or the inside of the tubes; the water-tube heater being the more efficient, due to the more rapid circulation of the heat absorbing medium in this type. On the above basis with exhaust steam at 212° F. and an average water temperature of 180° F. we would have transmission efficiencies of $(212 - 180) \times 180 = 5760$ B.t.u. and $(212 - 180) \times 270 = 8640$ B.t.u. per sq. ft. of surface per hour, respectively.

If rates of 5000 and 8000 B.t.u., respectively, are allowed for the two types of heaters and the transmission rate for the water radiation is taken at 200 B.t.u. per sq. ft. per hour, we have the following ratios between tube and radiating surface: 1 to 25 and 1 to 40, for the two conditions.

Since commercial feed-water heaters of the closed type are ordinarily rated on one-third of a sq. ft. of tube surface per horsepower, it is often customary to rate exhaust heaters in horsepower, or $3 \times$ sq. ft. of tube surface = hp. rating.

The live steam heaters will have a transmission rate varying with the temperature of the steam, which is usually at boiler pressure, and with the average temperature of the water, the maximum temperature of which may run as high as 250° F. to overcome extreme weather conditions. The tube surface in the high-pressure heater is usually one-third to one-half that in the exhaust steam heater.

Since the heaters are practically tubular boilers standing on end, standard practice in boiler design can be followed in determining thickness of shells, heads, rivet spacing, flanges, etc.

Heater Connections. The *steam connections* at the exhaust steam heater are proportioned on a velocity of 6000 ft. per minute, and a volume of 26 cu. ft. per pound at atmospheric pressure. $A = \frac{S \times 26}{6000}$, where A = area in sq. ft. of exhaust steam connection, and S = pounds of steam per minute. $S \times 850 = W \times 30$ or $S = \frac{30}{850} W = \frac{1}{28} \times W$, where W = weight of water in pounds per minute, heated from 150° to 180° F., and 850 is taken as the heating value of exhaust steam in B.t.u. See also the Chapter on "Exhaust Steam Heating."

The steam connections at the live steam heater may be determined in a similar manner.

The *water connections*, flow and return, are made the full size of the water mains, and these connections carried full size to pumps and through all by-passes.

Size of Water Heating Boiler. Water heating boilers are sometimes arranged as auxiliaries to the exhaust heaters, and one or more boilers of a battery may be so connected that the circulating water can be sent through them before leaving the plant. The amount of water to be circulated per boiler horsepower (b.hp.) can be determined as follows: 1 b.hp. $= 34.5 \times 970 = 33{,}465$ B.t.u. per hour. Also 1 sq. ft. of water radiation transmits 170 B.t.u. per hour, and allowing 10 per cent for line losses, we have 1 b.hp. $= \frac{33{,}465}{1.10 \times 170} = 180$ sq. ft. water radiation.

If each pound of water is reheated from 150° to 180° F. we have $33{,}465/30 = 1115$ lb. of water per hour can be circulated per b.hp. Hence, in this case, the boiler horsepower required is $M = \frac{60 \times W}{1115}$, where W = pounds of water circulated per min.

Circulating Pumps. The circulating pumps are usually of the centrifugal, single stage type, having an average operating efficiency of 70 per cent against heads up to 125 feet.

It is generally advisable to install the pumps in duplicate, to provide for contingencies and

insure continuous operation. In such cases each pump may be made equal to two-thirds of the maximum capacity required.

The *size of pumps* for large installations is generally limited to a capacity of 200 cu. ft. of water per minute or sufficient to supply 75,000 sq. ft. of radiation. These pumps are installed in parallel so that additions to the system may be readily made if desired.

Steam turbines are generally used for driving these pumps, although electric motors are sometimes employed. The steam consumption of these units is of little consequence as the exhaust is always utilized in the exhaust heater.

Centrifugal Pump Capacities. The two tables following, by *Charles L. Hubbard,* may be referred to in selecting size of pumps, speed of same, size of discharge and suction, and water horsepower required.

Table 13 applies to low-speed pumps adapted to direct-connected engine drive, while Table 14 is for higher speeds, with pump direct connected to steam turbine or motor.

TABLE 13

CAPACITIES, SPEEDS, AND HORSEPOWERS OF CENTRIFUGAL PUMPS OF VARIOUS SIZES AND FRICTION HEADS FROM 16 TO 90 FEET

Size of Delivery Inches	Rated Quantity of Discharge, in Gallons per Minute	Friction Head in Feet and Revolutions per Minute											Diameter of Impeller, Inches	Horsepower for Each Foot of Lift
		16 Ft.	20 Ft.	25 Ft.	30 Ft.	35 Ft.	40 Ft.	50 Ft.	60 Ft.	70 Ft.	80 Ft.	90 Ft.		
2	100	460	510	570	620	670	710	795	870	940	1,000	1,060	18	0.063
3	242	380	420	470	510	550	580	645	700	750	800	850	22	.136
4	430	310	340	370	405	435	465	515	560	600	640	670	26	.217
5	734	279	295	320	350	370	400	435	470	510	545	570	29	.309
6	1,050	240	265	290	320	340	360	395	435	465	490	520	32	.446
7	1,439	220	250	275	300	320	340	375	405	435	465	490	34	.606
8	1,880	210	235	260	280	300	320	355	390	415	440	465	36	.791

(*C. L. Hubbard*)

Referring to Table 13, it is seen that a pump having a 2-inch delivery and an 18-inch impeller will deliver 100 gallons per minute against a 40-foot head, when running at a speed of 710 r.p.m., and will require $0.063 \times 40 = 2.5$ horsepower for driving it.

TABLE 14

CAPACITIES, SPEEDS, AND HORSEPOWERS OF CENTRIFUGAL PUMPS OF VARIOUS SIZES FOR FRICTION HEADS FROM 4 TO 40 FEET

Size of Delivery, Inches	Rated Quantity of Discharge, in Gallons per Minute	Friction Head in Feet and Revolutions per Minute											Diameter of Impeller, Inches	Horsepower for Each Foot of Lift
		4 Ft.	6 Ft.	8 Ft.	10 Ft.	12 Ft.	16 Ft.	20 Ft.	25 Ft.	30 Ft.	35 Ft.	40 Ft.		
2	100	570	690	790	860	930	1,060	1,190	1,310	1,430	1,535	1,635	8	0.063
3	242	500	600	680	745	810	920	1,020	1,130	1,220	1,320	1,400	9	.136
4	430	440	510	580	645	695	790	875	970	1,050	1,130	1,195	10	.217
5	734	370	430	525	570	615	690	765	840	910	970	1,030	11	.309
6	1,058	330	400	480	530	570	650	710	780	845	905	960	12	.446
7	1,439	320	370	420	450	490	550	605	670	720	775	820	14	.606
8	1,880	280	320	360	395	425	480	525	580	630	675	715	16	.791

(*C. L. Hubbard.*)

NOTE.—These tables are based on overall efficiencies of 40 per cent for low heads up to 60 per cent for high heads. That is, from 40 per cent to 60 per cent of the power supplied the engine, turbine or motor driving pump represents the water horsepower output of the pump.

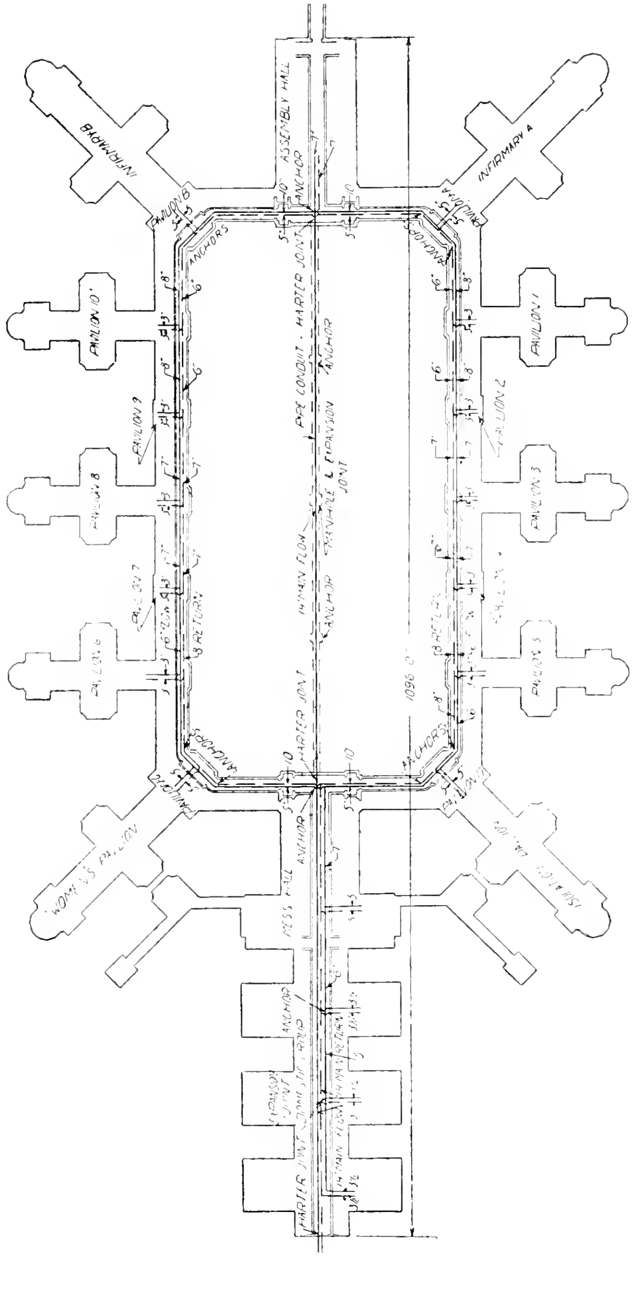


Fig. 26. Plan of Hospital, Showing Arrangement of Main Heating Pipes. (*Practical Engineer.*)

Expansion Tanks. It is necessary to provide, as in a gravity circulating system, an expansion tank with a capacity of from 2 to 5 per cent of the total volume of water in the system. This tank may be of the open or closed type, depending on the maximum water temperature to be used, as temperatures above 212° F. require a closed tank.

The tank is generally located in the plant, although it may be placed in the highest building on the line. The closed tank is generally placed in the plant and connected to the return main on the suction side of the pumps. Closed tanks should have a 2″ safety valve with waste pipe,

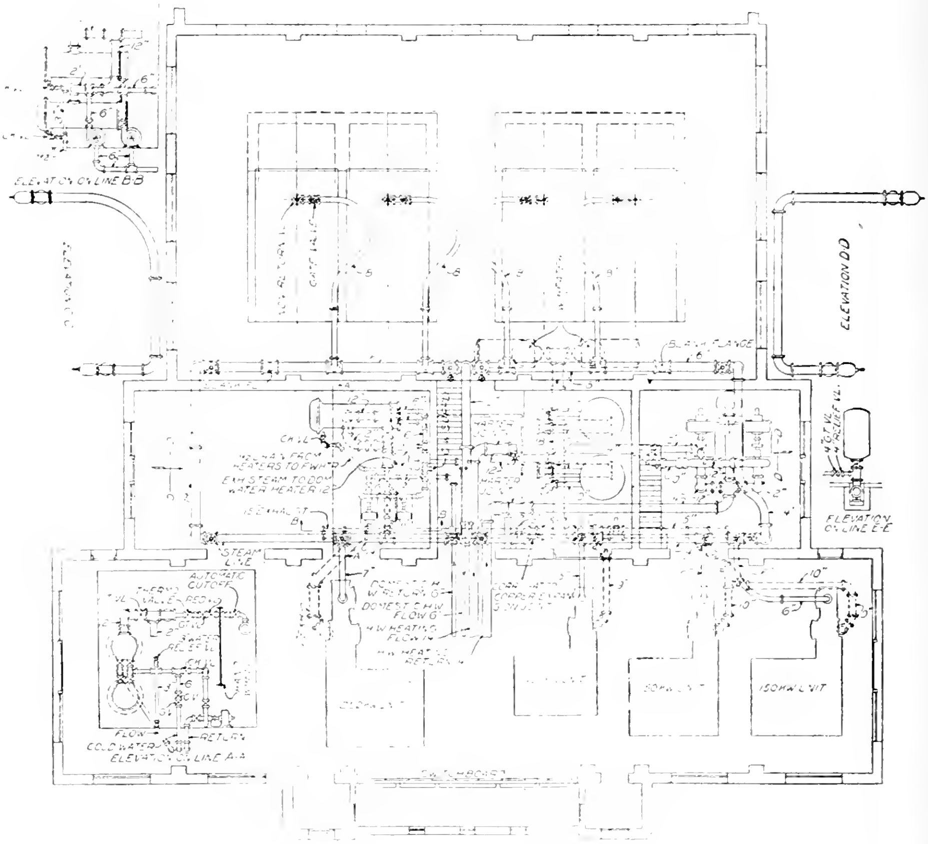


FIG. 27. GENERAL PLAN AND PARTIAL ELEVATIONS OF THE POWER PLANT. (*Practical Engineer.*)

and must have some means of feeding water or air at a pressure of from 3 to 5 lb. above the static head on the system.

The volume of water in the system can be estimated on the basis of 1½ pints per sq. ft. of cast-iron radiation and 1 pint per sq. ft. of pressed steel or pipe coil radiators. Add 100 per cent for water in building mains and risers, and calculate volume in distributing system to get total volume of water to be allowed for.

Accessories for Central Hot-Water Heating Equipment. The system should be equipped with *pressure gages* on the supply and return to each pump, and on the mains where they leave the plant.

Thermometers should be installed on the flow and return connections to each heater, and on the flow and return mains where they leave the plant. The two latter thermometers should be of the recording type, and a recording outside air thermometer should also be installed.

Anemometers are of value in estimating the weather conditions to be overcome, and the plant should be equipped with such an apparatus, or some other type of wind gage to determine the wind velocity.

TYPICAL CENTRAL HOT-WATER HEATING SYSTEM

The following description of the equipment installed in the plant of the Lima State Hospital, at Lima, Ohio, is taken in part from the "Practical Engineer" for June 1, 1914. This plant is a combination heat, light, and power plant serving a fairly compact group of a dozen buildings (Fig. 26).

Boiler Plant. There are four McNaul water-tube boilers of the four-pass type of 400 rated hp. each, operating at 150 lb. gage pressure and supplying steam to one 250-kw., one 100-kw., and two 150-kw. units (Fig. 27).

Ohio pea, nut, or slack coal is used as fuel and is fed to furnaces by Jones underfeed stokers, supplied with forced draft if necessary. The flue gases are discharged to a brick chimney 204 ft. high by 10 ft. in diameter at base, tapering to 6 ft. at the top.

The feed water is heated by two Sims open heaters of 2000 hp. capacity each, which receive the condensate from the exhaust and live steam hot-water heaters, and high pressure drips.

Hot-Water Heating System. A two-pipe main system is used with a reversed return system so that equal length of water travel is secured to all radiators. This is accomplished by running the 14-inch flow main (Fig. 26) clear to the far end of the group of buildings before taking off any branches. This main then divides as it runs around the court and decreases in size, while the return mains begin at the far end and increase in size as they continue around the court and back to the plant, running parallel to the two branches of the flow main, until they join in the single 14-inch return main.

The circulation through the heating system is produced by two Weinman centrifugal pumps, which take water at 25-lb. pressure from the 14-inch return and discharge it at 40-lb. pressure into the 14-inch line to the heaters.

There are four heaters, all of the closed type, two operating on exhaust and two on live steam. The live steam heaters are in series with the exhaust heaters, but in parallel with each other.

The circulating pumps are driven by Wood 230-volt, direct-current motors of 110 hp., with speed variable from 660 up to 1000 r.p.m. by means of a Fort Wayne controlling system, and each pump has a maximum capacity of 3000 gal. per minute. To avoid any excess pressure in the system, an expansion tank, shown in the view of the pump room, Fig. 27, is connected to the suction side of the pumps, and is set to open the relief valve at 60 lb. pressure. Throughout the plant, the different pipes are distinguished by the coloring, the return pipes on the heating system being a terra cotta, and the outflow pipes yellow.

The heaters are termed converters, those for the exhaust heating having steel shells and tubes, each heater having a capacity for heating 1500 gal. per minute from 170 deg. up to 200 deg., using exhaust steam at atmospheric pressure. Each heater contains 540 tubes, and weighs 36,500 lb. These heaters are set vertically, with steam connections as shown in Fig. 28.

In front of these are located the two high-pressure converters, which are horizontal type, each containing 270 brass tubes, 1½ in. outside diameter, having one tube head fixed and the other floating. These are of the multi-pass type, the water passing six times the length of the heater between inlet and outlet, through 45 tubes for each pass. The capacity of each heater

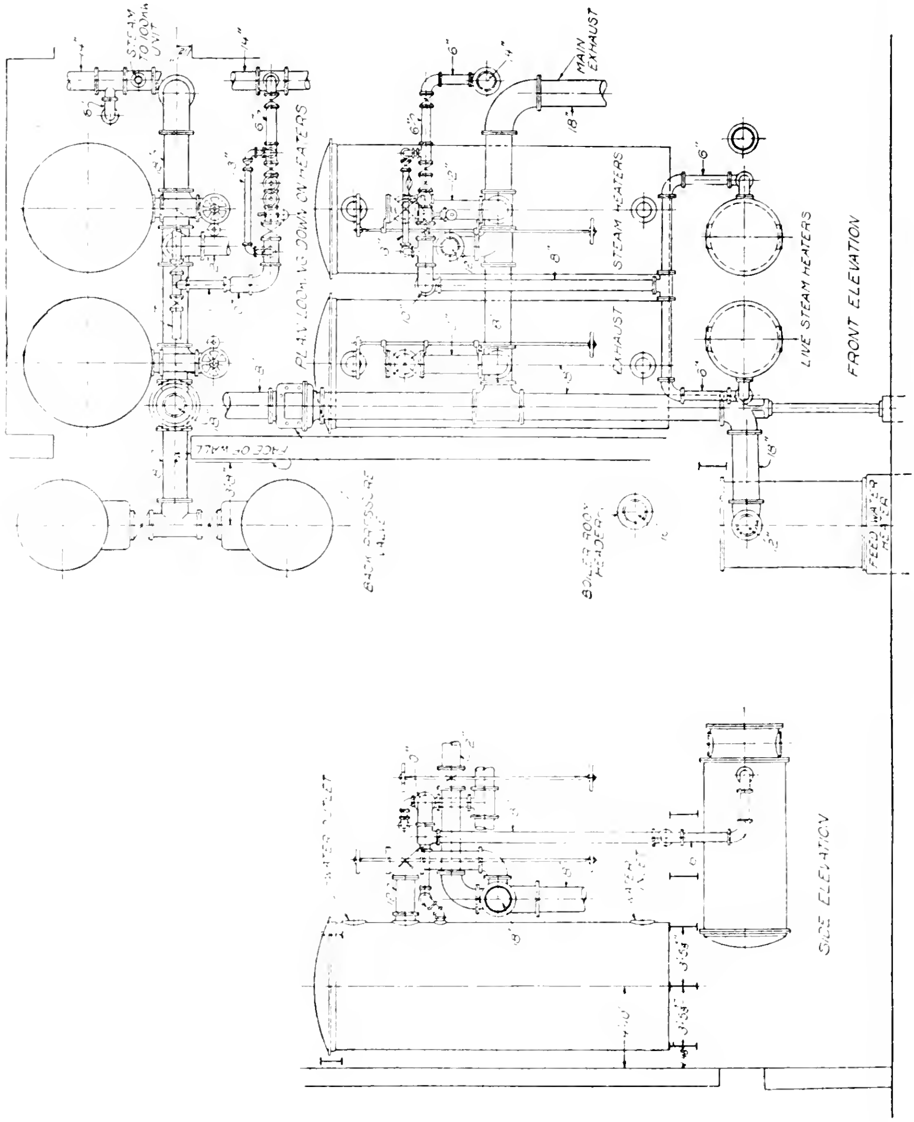


FIG. 28. STEAM CONNECTIONS OF THE HOT-WATER HEATERS.
(*Practical Engineer.*)

is 1500 gal. per min., heating from 180 deg. up to 210 deg., using steam at 40-lb. pressure, this steam being reduced from the boiler pressure of 150 lb. by means of a Keiley reducing valve.

The heaters are built to give rapid flow through the multi-pass, in order to accomplish quick heat transfer from the steam to the water, and also to keep the tubes of the heaters clean. The tube plate covers are divided by partitions into sections to give the several passes, and are so arranged that they can be removed, leaving the tubes free for cleaning or inspection. The water connections are shown in Fig. 29.

The high-pressure heaters are not used so long as the exhaust steam is sufficient to give

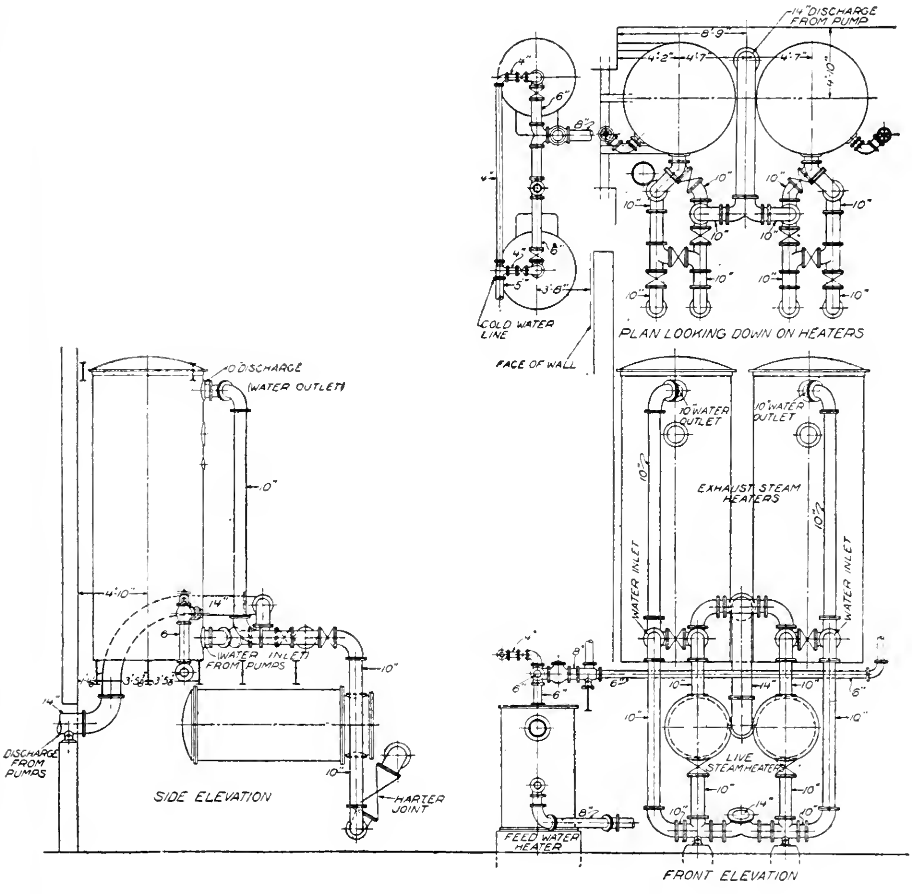


FIG. 29. WATER CONNECTIONS OF THE HOT-WATER HEATERS. (*Practical Engineer.*)

the temperature required for heating the building, but if the temperature of the outflow water falls below that necessary, the high-pressure heaters are automatically cut in, in series with the exhaust. If given a horsepower rating, the high-pressure heaters would be 4,000 hp. each, but this has no particular significance in the present plant.

Leaving the heaters, the water passes to the flow main through a Harter expansion joint

and skew fitting and enters the 14-inch flow main. This main enters the 500-ft. tunnel, which is 8 ft. high and 10 ft. wide, passing through the basement of the hospital building and then under the court to the far end, where it branches in either direction, running back toward the plant in the basement corridors around the court as already explained.

The Hot- and Cold-Water Supply System. The hot water for domestic purposes is heated in a room to one side of the pump room, by means of two Sims closed heaters of horizontal type, shown on the plan of the power plant (Fig. 27). Each heater has 102 brass tubes, with one fixed and one floating tube head, and is divided into six passes, having 17 tubes each, the capacity being 300 gal. per min., raising the temperature from 50 up to 180 deg. F., with steam at 40-lb. pressure. These heaters use live steam entirely, as the amount required is not large, and during the summer months, when little exhaust is required for heating the building, the remainder will be necessary for heating the feed-water for boilers.

Circulation in the domestic hot-water system is provided by two duplicate No. 6 Weinman centrifugal pumps, each having a capacity of 500 gal. per min., and driven by a 25-hp. Wood motor, with speed variable, by Fort Wayne field and armature control, from 1200 to 1800 r.p.m. The flow and return lines for this system are carried through the tunnel and basement corridors near the heating lines. Live steam is furnished to these heaters from the engine header through a Keiley reducing valve.

The *cold-water system*, which furnishes make-up water for the boilers, for the forced hot-water system, heating system, and for domestic service, is supplied from a 100,000-gal. overhead tank and stand-pipe carried on a structural steel tower. This tank gives 128 ft. head, or practically 50-lb. pressure on the water system, and is kept filled by pumps which draw their supply from a 3,500,000-gal. reservoir located some 500 ft. from the power plant. Two Weinman centrifugals are used, having a maximum lift of 23 ft. and a capacity of 250 gal. per min., at 1600 r.p.m., the suction pipe being 6 in., and the discharge pipe from each pump 3 in. These pumps are driven by Northern 20-hp. interpole motors, which are controlled by the head in the stand-pipe, and will operate on a drop of 3 ft. head.

Discharge from pumps is connected to the stand-pipe bottom, to the domestic heater, to the flow water and the boiler feed-water heater, a check valve being placed between the stand-pipe and the other parts of the system to prevent return flow from any part of the system into the stand-pipe, in case of a drop in stand-pipe pressure.

CHAPTER XX

PIPE, FITTINGS, VALVES, COVERINGS AND ACCESSORIES

PIPE

Commercial Classification of Pipe. Commercial pipe is made of *wrought-iron* or *mild steel*, in certain definite sizes, always stated in terms of the nominal internal diameters up to and including 12″. (Table 1.) Above 12″ internal diameter the size is based on the outside diameter, and the thickness of metal always specified.

There are three *weights* or *strengths* of pipe generally recognized in engineering practice, known as "standard," "extra strong" and "double extra strong," all of which have the same outside diameter for a given size.

Standard Pipe. Standard pipe is also known as *full weight* pipe and is made from sheets of sufficient thickness to permit of the necessary manipulation, such as heating and rolling, and still finish in random lengths of from 18 to 20 ft. which will weigh, including coupling on one end, within 5 per cent of "card weight" (Table 1). Unless otherwise specified, this pipe is furnished in random lengths with threads and couplings.

TABLE 1

DIMENSIONS OF STANDARD AND EXTRA STRONG* WROUGHT-IRON AND STEEL PIPE

Nominal Size	Diameter			Circumference			Internal Transverse Area		Length of Pipe in Ft. per Square Ft. of Exter'l Surface	Nominal Weight Lb. per Foot	
	External Standard and Extra Strong	Internal Standard	Internal Extra Strong	External Standard and Extra Strong	Internal Standard	Internal Extra Strong	Standard	Extra Strong		Standard	Extra Strong
⅛....	0.405	0.269	0.215	1.272	0.848	0.675	0.0573	0.0363	9.440	0.244	0.314
¼....	.540	.364	.302	1.696	1.144	.949	.1041	.0716	7.075	.424	.535
⅜....	.675	.493	.423	2.121	1.552	1.329	.1917	.1405	5.657	.567	.738
½....	.840	.622	.546	2.639	1.957	1.715	.3048	.2341	4.547	.850	1.087
¾....	1.050	.824	.742	3.299	2.589	2.331	.5333	.4324	3.637	1.130	1.473
1	1.315	1.049	.957	4.131	3.292	3.007	.8626	.7193	2.904	1.678	2.171
1¼....	1.660	1.380	1.278	5.215	4.335	4.015	1.496	1.287	2.301	2.272	2.996
1½....	1.900	1.610	1.500	5.969	5.061	4.712	2.038	1.767	2.010	2.717	3.631
2	2.375	2.067	1.939	7.461	6.494	6.092	3.356	2.953	1.608	3.652	5.022
2½....	2.875	2.469	2.323	9.032	7.753	7.298	4.784	4.238	1.328	5.793	7.661
3	3.500	3.068	2.900	10.996	9.636	9.111	7.388	6.605	1.091	7.575	10.252
3½....	4.000	3.548	3.364	12.566	11.146	10.568	9.887	8.888	0.955	9.109	12.505
4	4.500	4.026	3.826	14.137	12.648	12.020	12.730	11.497	.849	10.790	14.983
4½....	5.000	4.506	4.290	15.708	14.162	13.477	15.961	14.454	.764	12.538	17.611
5	5.563	5.047	4.813	17.477	15.849	15.121	19.990	18.194	.687	14.617	20.778
6	6.625	6.065	5.761	20.813	19.054	18.099	28.888	26.067	.577	18.974	28.573
7	7.625	7.023	6.625	23.955	22.063	20.813	38.738	34.472	.501	23.544	38.048
8	8.625	7.981	7.625	27.096	25.076	23.955	50.040	45.664	.443	28.544	43.388
9	9.625	8.941	8.625	30.238	28.089	27.096	62.776	58.426	.397	33.907	48.728
10	10.750	10.020	9.750	33.772	31.477	30.631	78.839	74.662	.355	40.483	54.735
11	11.750	11.000	10.750	36.914	34.558	33.772	95.033	90.763	.325	45.557	60.075
12	12.750	12.000	11.750	40.055	37.700	36.914	113.098	108.43	.299	49.562	65.415

NOTE.—Dimensions are nominal and, except where noted, are in inches.

* Often called extra heavy pipe.

A lighter weight of standard pipe, in sizes up to 6″, known as *merchant pipe*, and running about 10 per cent below "card weight" has been discontinued by the principal manufacturers. Unless this pipe is wanted, it is necessary to specify "full weight" pipe.

Extra Strong Pipe. Extra strong pipe (Table 1) is usually specified for steam, gas or hydraulic work at pressures above 125 lb. gage. This pipe is made in random lengths of from 12 to 20 ft. and is always furnished with plain ends unless otherwise specified, although as much as 10 per cent of a total order may be in lengths from 6 to 12 ft.

Double extra strong pipe is omitted from Table 1 since its use is limited almost entirely to high pressure hydraulic work. The same trade practice is followed in furnishing it as for extra strong pipe.

Outside Diameter Pipe. Outside diameter pipe, known as O. D. pipe (Table 1*a*) is the commercial designation applied to all regular sizes above 12″. Since the terms standard or extra strong do not apply to these sizes, it is always necessary to give the thickness as well as the outside diameter. This pipe is furnished in random lengths of from 8 to 20 ft., depending on the size, and with plain ends. The threading of O. D. pipe is not recommended.

TABLE 1*a*

OUTSIDE DIAMETER (O. D.) STEEL PIPE

Nominal weight in pounds per foot

Size Outside Diam.	Thickness								
	¼ In.	5/16 In.	⅜ In.	7/16 In.	½ In.	9/16 In.	⅝ In.	11/16 In.	¾ In.
14........	36.75	45.72	54.61	63.42	72.16	80.80	89.36	97.84	106.20
15........	39.42	49.06	58.62	68.10	77.50	86.81	96.03	105.20	114.20
16........	42.09	52.40	62.63	72.78	82.85	92.83	102.70	112.50	122.20
17........	44.76	55.74	66.64	77.46	88.19	98.84	109.40	119.90	130.30
18........	47.44	59.08	70.65	82.14	93.54	104.80	116.10	127.20	138.30
20........	52.78	65.76	78.67	91.49	104.20	116.90	129.40	141.90	154.30
21........	55.45	69.10	82.68	96.17	109.60	122.90	136.10	149.30	162.30
22........	.....	72.44	86.68	100.80	114.90	128.90	142.80	156.60	170.30
24........	.....	79.13	94.70	110.20	125.60	140.90	156.20	171.30	186.30
26........	.....	.....	102.70	119.50	136.30	152.90	169.50	186.00	202.40
28........	.....	.....	110.70	128.90	147.00	165.00	182.90	200.70	218.40
30........	.....	.....	......	138.20	157.70	177.00	196.30	215.40	234.40

In connection with pipe sizes, Table 2, giving certain tube data, may be found to be of service.

TABLE 2

TUBE DATA, STANDARD OPEN-HEARTH OR LAP-WELDED STEEL TUBES

Size Extern. Diam.	B. W. Gage	Thickness	Internal Diam.	Circumference		Transverse Area Square Inches		Square Feet of External Surface per Ft. of Length	Length in Feet per Sq. Foot of External Surface	Nominal Weight Pounds per Ft.
				External	Internal	External	Internal			
1½....	10	.134	1.232	4.712	3.870	1.7671	1.1921	.392	2.546	1.955
1½....	9	.148	1.204	4.712	3.782	1.7671	1.1385	.392	2.546	2.137
1½....	8	.165	1.170	4.712	3.676	1.7671	1.0751	.392	2.546	2.353
2	10	.134	1.732	6.283	5.441	3.1416	2.3560	.523	1.909	2.670
2	9	.148	1.704	6.283	5.353	3.1416	2.2778	.523	1.909	2.927
2	8	.165	1.670	6.283	5.246	3.1416	2.1904	.523	1.909	3.234
3¼....	11	.120	3.010	10.210	9.456	8.2958	7.1157	.850	1.175	4.011
3¼....	10	.134	2.982	10.210	9.368	8.2958	6.9840	.850	1.175	4.459
3¼....	9	.148	2.954	10.210	9.280	8.2958	6.8535	.850	1.175	4.903
4	10	.134	3.732	12.566	11.724	12.566	10.939	1.047	.954	5.532
4	9	.148	3.704	12.566	11.636	12.566	10.775	1.047	.954	6.000
4	8	.165	3.670	12.566	11.530	12.566	10.578	1.047	.954	6.758

NOTE.—Dimensions are nominal and, except where noted, are in inches.

Threading Pipe. The threading of either wrought-iron or steel pipe requires suitable dies adapted to the metal to be cut. Dies suitable for wrought iron will tear steel pipe, and hence the complaint is sometimes made that steel pipe is brittle. This can be readily overcome by using proper dies. All pipe is threaded uniformly using *Briggs* standard gage and taper. This taper of ¾″ to 1′–0″ on all standard pipe threads is necessary in order to secure a tight joint in the threads when screwing the pipe into a fitting or valve.

Testing Pipe. Pressure tests at the mill of wrought-iron or steel pipe are commonly made in order to show the presence of flaws or other defects in the *weld* or body of the pipe. Wrought pipe, as distinguished from seamless tubing, is either *butt* or *lap-welded;* sizes up to and including 1¼″ being made by the former, and those 1½″ and larger by the latter process. Lap-welded pipe, 1½″ diameter, may safely be tested to 2500 lb. per sq. in. cold hydraulic pressure, while 12″ diameter pipe should not be tested to more than 300 lb. per sq. in. The makers vary the test pressure in accordance with the diameter so as to produce approximately the same fiber stress in each size of pipe.

The *theoretical bursting pressures* for steel pipe of varying diameters ranging from ⅛″ diameter to 12″ diameter can be calculated, and are given by *John B. Berryman* in Table 3.

TABLE 3

THEORETICAL BURSTING PRESSURE OF WROUGHT-IRON PIPE

Based on New Material with Plain Ends. Weld Assumed to Be Perfect

(Full weight standard pipe)

Size, Inches	Bursting Pressure, Pounds	Working Pressure Factor of Safety 6, Pounds	Size, Inches	Bursting Pressure, Pounds	Working Pressure Factor of Safety 6, Pounds
⅛	20,142	3,357	3½	5,100	850
¼	19,338	3,223	4	4,704	784
⅜	14,730	2,455	4½	4,350	725
½	13,992	2,332	5	4,104	684
¾	10,968	1,828	6	3,690	615
1	10,224	1,704	7	3,426	571
1¼	8,112	1,352	8	3,228	538
1½	7,200	1,200	9	3,078	513
2	5,958	993	10	2,922	487
2½	6,612	1,102	12	2,496	416
3	5,658	943			

The generally accepted formula for bursting pressure of a cylinder is:

$$P = 2 \times \frac{t \times S}{D}$$

in which, P = bursting pressure in pounds per sq. in.

t = thickness of metal in inches.

S = tensile strength in pounds per sq. in. = 40,000 for wrought iron and 50,000 for steel.

D = pipe diameter in inches.

Example. Find the bursting pressure of 10″ full weight steel pipe, 0.366″ thick, and 10.019″ actual internal diameter.

$$P = \frac{2 \times 0.366 \times 50{,}000}{10.019} = 3{,}653 \text{ lb. per sq. in.}$$

If we wish to find the proper thickness of metal it is only necessary to solve the equation above for t and we have

$$t = \frac{D \times P}{2 \times S}.$$

Apparent Factor of Safety. The proper factor of safety to be employed is a matter of judgment, but for steam piping it should never be less than six. In steam lines there are stresses due to vibration, expansion and contraction, and possible shock. In water lines there may be severe shocks due to water hammer or the less severe but continual shocks from the action of the pumps. The element of corrosion has also to be considered as in some cases the original thickness of the metal may be reduced one-half in a comparatively short time. As these disturbing elements can only be assumed, it is evident that factors of safety from eight to fifteen may be employed with advantage.

The *Crane Co.* has made some bursting tests on 10-inch pipe, with the following results:

10-inch standard wrought iron, burst 1900 lb., by rule 2922 lb.

10-inch standard steel, burst 3000 lb., by rule 3648 lb.

10-inch extra strong wrought iron, burst 2700 lb., by rule 4102 lb.

None of the pieces destroyed burst at the weld, the rupture in each case being some distance from it.

Specifications for Pipe. Specifications for wrought-iron and steel pipe for the usual service conditions existing in steam systems are given in the chapter on "Power Plant Piping."* In general, three classes of service are recognized as follows: (1) Service where pressures are 125 lb. per sq. in. or less; (2) where pressures are above 125 lb. but less than 250 lb., and (3) where pressures are less than 250 lb. but the steam is superheated. In other words, the piping as well as all valves and fittings for steam service must be adopted to either "low-pressure," "high-pressure," or "superheated" service.

EXPANSION OF PIPING

Determination of Expansion. Expansion of piping is ordinarily based on the theoretical elongation of the measured length of line for the difference in temperature between the air at the time the pipe was fitted and the final temperature when filled with steam, hot water, or gas. This elongation depends on the coefficient of linear expansion, which for steel is 0.00067 per 100° F. per 1′ 0″, or for a line 500 ft. long, fitted on a zero day and intended for steam service at 212° F., we would have $\frac{12 \times 500 \times 0.00067 \times 212}{100} = 8.5''$ increase in length. The following table is computed in this manner for steel pipe for varying temperatures and pressures:

TABLE 4

EXPANSION OF WROUGHT-IRON AND STEEL PIPE

Temperature F.°	Gage Pressure Pounds per Square Inch	Linear Expansion in Inches
160	Hot water	1.02
200		1.34
212	0	1.43
240	10	1.66
259	20	1.81
274	30	1.94
297	50	2.12
338	100	2.70
365	150	3.05
388	200	3.31
422	300	3.73
500		4.76
600	Superheated steam	6.23
650		7.03

NOTE.—Column 3 gives the theoretical increase in length of 100 feet of pipe when heated from 32° F. to the temperature or pressure given in the table. Expansion is stated in inches.

*See Volume II.

In general, the amount of lineal expansion E in inches per 100 ft. of length of pipe of any material may be determined by the following equation, using proper value for C, or from Fig. 1:

$$E = 1200 \times C \times (T - t).$$

E = expansion in inches per 100 ft. of pipe.

$(T - t)$ = temperature difference in degrees Fahrenheit.

C = coefficient of lineal expansion.
= 0.0000111 for bronze.
= 0.0000105 for drawn brass.
= 0.0000095 for copper or cast brass.
= 0.0000068 for wrought iron.
= 0.0000067 for steel.
= 0.0000065 for cast iron.

Methods of Providing for Expansion. The proper *provision for the expansion and contraction of piping* must be made in all cases where water, steam, or gas is to be used at high temperatures, and is usually accomplished by *long sweep bends* or *expansion joints*. Certain points, usually where branches are taken off, are securely anchored to the building structure, and the movement between these points taken up by the expansion members, such as bends or joints.

Dimensions of Pipe Bends. The allowable dimensions for pipe bends are limited by the practical considerations involved in actually bending the pipe, and Table 5 will serve as a guide in laying out expansion bends.

The radius of any bend made from pipe $2\frac{1}{2}$ in. and larger should not be less than five diameters of the pipe, and a larger radius is much preferable. When bends are used to take up expansion, the longer the radius the better. The figures in the following table apply to all forms of bends, and show what dimensions are required by the mill if bends are to be made of proper proportion:

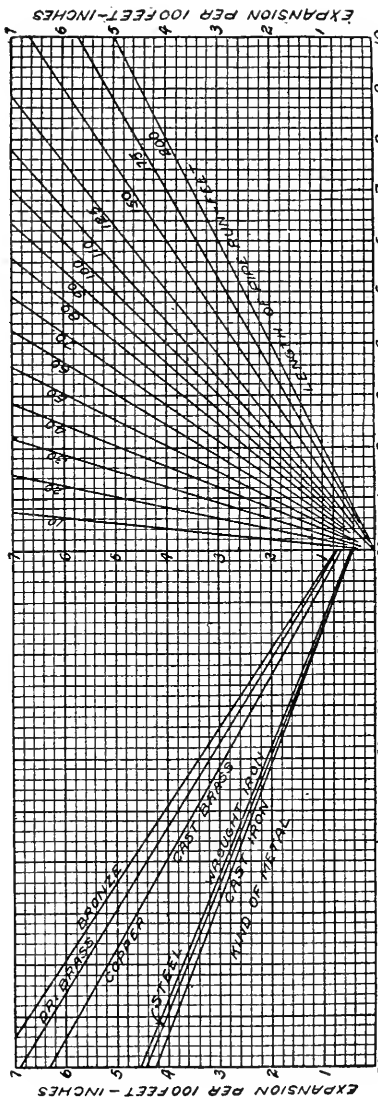


Fig. 1. Lineal Expansion of Pipes of Various Materials for Different Temperature Changes and Lengths.

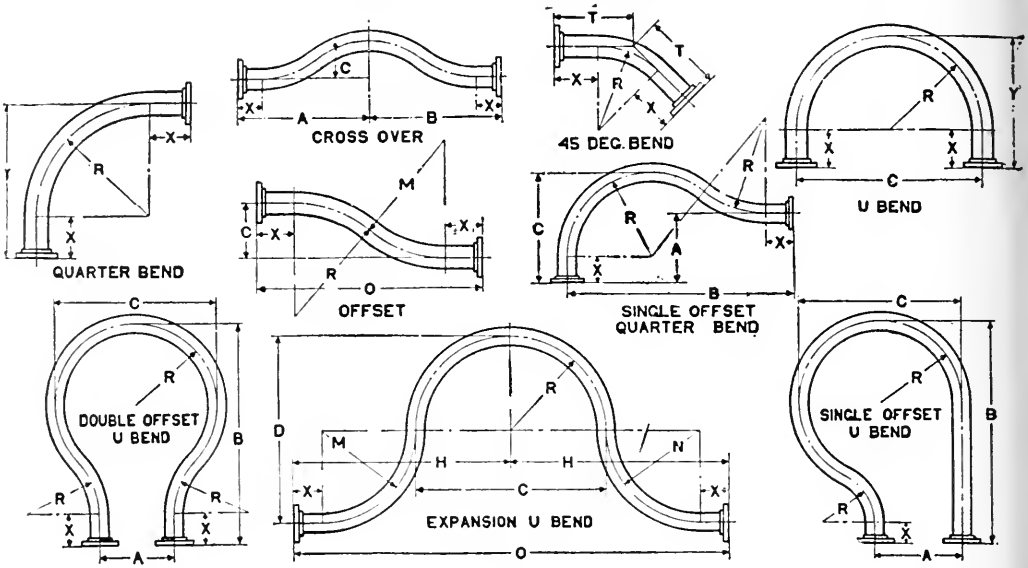


FIG. 2.

TABLE 5

PIPE BENDS MADE FROM LAP-WELDED STEEL PIPE

(See Fig. 2)

Size of Pipe	R-M-N Advisable Radius of Bends	*T* Center to End or Face of Flanges		*X* Length of Tangents or Straight Pipe on Each Bend	*Y* Center of Bends to Face of Flanges or Ends of Pipe		Lineal Feet of Pipe in Each Quarter Bend		Lineal Feet of Pipe in Each U-Bend		Lineal Feet of Pipe in Each 45 Deg. Bend		Minimum Radius to Which Bends can Be Made from Extra Strong Pipe Only, Inches
Inches	Inches	Feet	Inches	Inches	Feet	Inches	Feet	Inches	Feet	Inches	Feet	Inches	
2½....	12½	0	9 3/16	4	1	4½	2	3¾	3	11¼	1	5⅞	7
3	15	0	10¼	4	1	7	2	7¾	4	7⅛	1	7⅞	8
3½....	17½	1	¼	5	1	10½	3	1½	5	5	1	11¾	10
4	20	1	1¼	5	2	1	3	5½	6	1	2	1¾	12
4½....	22½	1	3 5/16	6	2	4½	3	11½	6	10¾	2	5¾	14
5	25	1	4⅜	6	2	7	4	3¼	7	6⅝	2	7½	15
6	30	1	7 7/16	7	3	1	5	1⅛	9	¼	3	1½	20
7	35	1	10½	8	3	7	5	11	10	6	3	7⅜	24
8	40	2	1 9/16	9	4	1	6	9	11	11¾	4	1½	28
9	45	2	5⅝	11	4	8	7	8¾	13	7⅜	4	9¼	35
10	50	2	8¾	12	5	2	8	6½	15	1⅛	5	3¼	40
12	60	3	2⅞	14	6	2	10	¼	18	¼	6	3¼	50
14	70	3	9	16	7	2	11	10	21	....	7	3	65
15	75	3	11 1/16	16	7	7	12	6	..	....	7	7	70
16	80	4	3⅛	18	8	2	13	5¾	..	....	8	2¾	78
18	108	5	2¾	18	10	6	17	1¾	..	....	10	⅞	88
20	120	5	7¾	18	11	6	18	8½	..	....	10	10¼	104
22	132	6	⅝	18	12	6	20	3	..	....	11	7¾	132
24	144	6	5⅝	18	13	6	21	10	..	....	12	⅝	144

Bends for *small pipe* may be made cold, within certain limits, as indicated in Table 5*a* by the *National Pipe Bending Co.*, although for minimum requirements this pipe must be bent hot as for large pipe.

TABLE 5*a*

DIMENSIONS OF MINIMUM BENDS AND COILS

Nominal Size of Pipe		Cold Bent											Hot Bent						
		$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	6	7	8
Least Ordinary	Center Radius 90° Bends.	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	4	6	8	10	18	20	22	24	27	30	36	42
	Center Diameter U-Bends	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	5	6	8	12	16	20	36	40	44	48	54	60	72	84
	Outside Diameter Coils...	5	6	7	8	10	12	15	18	20	24	42	44	49	54	60	72	84	96
Difficult	Center Radius 90° Bends.	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4	6	8	15	14	16	18	20	25	30	36
	Center Diameter U-Bends	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	4	5	6	8	12	16	30	28	32	36	40	50	60	72
	Outside Diameter Coils...	3	4	5	6	8	10	12	14	16	20	36	32	36	42	48	60	72	84
Approximate Limit Varying with Circumstances	Center Radius 90° Bends.	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{4}$	3	4	5	6	10	12	14	16	20	25	30
	Center Diameter U-Bends	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	$4\frac{1}{2}$	6	8	10	12	20	24	28	32	40	50	60
	Outside Diameter Coils...	2	$2\frac{1}{2}$	3	4	6	7	8	10	12	14	16	24	30	36	42	48	60	72

NOTE.—Ends on bends should be straight for a length equal to the diameter of the pipe.

Expansion Allowed by Bends. The amount of expansion allowed for by bends depends upon the radius of the bend, increasing with it, and varies inversely as the thickness of the wall.

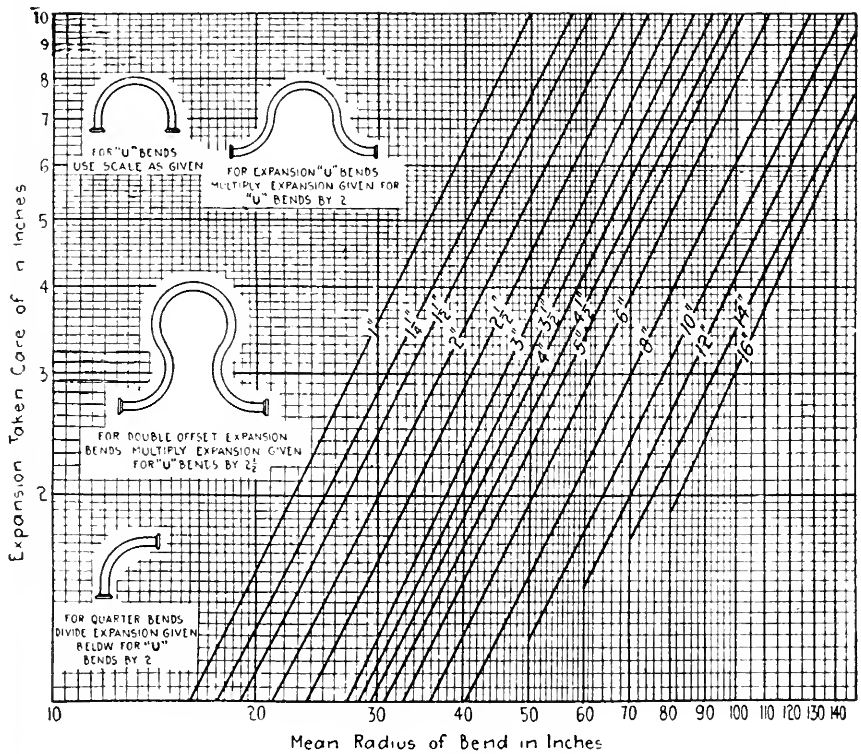


FIG. 3. EXPANSION ANY BEND WILL ALLOW WITHOUT INJURY.

Actual tests by the *Crane Co.* have been made on pipe bends made of pipe from 1″ to 16″ diameter, and the results have been plotted by *W. L. Durand*, and are given in Fig. 3.

The formula for U-bends is:

$$E = \frac{0.0052\, R^2}{d}$$

in which, E = expansion in inches.
R = mean radius of bend in inches.
d = outside diameter of pipe in inches.

If any two of the three values are given, the third can be easily found from the curves.

Example. What is the necessary radius for a U-bend to take care of 3 in. of expansion in an 8-in. pipe? Referring to the curves and running out horizontally from 3 in. to the line marked 8 in., the radius of the bend is read as 70 in.

Expansion Joints. Either single (Fig. 4) or double slip expansion joints, or corrugated copper expansion joints (Fig. 5), may also be used in lines when bends and offsets are not practicable.

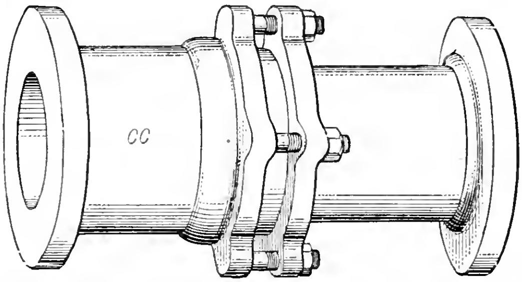


FIG. 4. STANDARD UNBALANCED SINGLE EXPANSION JOINT.

The allowable *traverse* or movement of these joints determines the number to be installed, or the lineal feet of pipe for which each joint may compensate. Joints of the single slip type are made up to allow a maximum traverse as follows (expressed in inches):

Pipe size........................	2 –2½–3 –3½–4 –4½–5–6–7–8–9–10–12
Traverse........................	2½–2½–2¾–3 –3½–3½–4–5–6–7–7– 7– 8

Joints may be specially made up to allow a *special traverse* of from 6 to 18 inches if desired, although it is generally customary to limit the traverse of one sleeve to from 3″ to 4″. Double slip

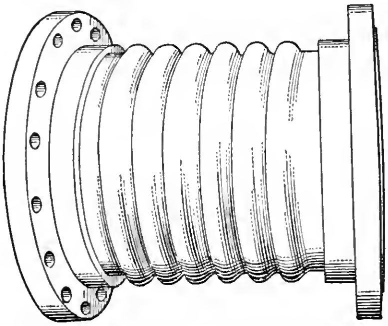


FIG. 5. CORRUGATED COPPER EXPANSION JOINT.

expansion joints are generally designed to allow a traverse of 4″ on each sleeve for pipe sizes from 2″ to 9″, and 3½″ on each sleeve pipe for sizes from 10″ to 16″.

Slip joints are usually made with iron bodies and brass sleeves, and must have an adjustable packing gland with follower (Fig. 4). Joints may be furnished *screwed* or *flanged* for standard or extra heavy service.

The actual and theoretical amount of expansion have been compared in a number of cases as it was formerly believed that piping would actually expand under steam temperatures about one-half the theoretical amount, due to the fact that the exterior of the pipe would not reach the full temperature of the steam in the pipe. It would appear, however, from recent experiments, that such actual expansion will in the case of well-covered pipe be very nearly the theoretical amount. In one case noted, a steam header, 293 feet long when heated under a working pressure of 190 pounds, the steam superheated approximately 125 degrees, expanded 8¾ inches; the theoretical amount of expansion under the conditions would be approximately $9\,{}^{35}/_{64}$ inches.

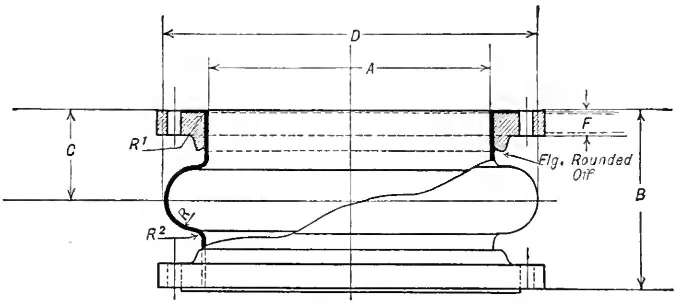
Heat loss from piping conveying steam or water should be prevented as completely as possible by the use of non-conducting coverings. The amount of this loss in uncovered lines and the saving that can be effected by insulated lines are considered later in this Chapter under "Coverings."

Bellows Type Copper Expansion Joints. The form of expansion joint shown in Table 6 is suitable for exhaust steam lines or for low-pressure water lines. These expansion joints are used almost universally on the connection between the exhaust outlet from a steam turbine and the exhaust inlet of its condenser.

The exhaust outlets of steam turbines as made at the present time are very seldom circular, but are more often oval or rectangular in shape.

TABLE 6

SIZES AND DIMENSIONS OF KELLOGG COPPER BELLOWS EXPANSION JOINT.



A Size, Inches	B Face to Face, Inches	C Center to Face, Inches	D Outside Diameter of Belt, Inches	R Radius of Belt, Inches	A Size, Inches	B Face to Face, Inches	C Center to Face, Inches	D Outside Diameter of Belt, Inches	R Radius of Belt, Inches
4	8	4	8¾	1½	24	14	7	31	2⅝
5	8	4	9	1½	26	14	7	34	2⅝
6	9	4½	10¾	1⅝	28	15	7½	36	3
7	9	4½	11¾	1⅝	30	15	7½	38	3
8	10	5	12¾	1⅝	32	16	8	40	3½
9	10	5	13½	1⅝	34	16	8	44	3½
10	11	5½	14½	1⅞	36	16	8	47	3½
12	11	5½	15¾	1⅞	38	17	8½	48	4½
14	12	6	20¼	2⅛	40	17	8½	50	4½
16	12	6	21¼	2⅛	42	17	8½	52	4½
18	13	6½	24½	2⅜	44	18	9	54	5
20	13	6½	26½	2¾	46	18	9	56	5
22	14	7	29	2⅝	48	18	9	58	5

Twenty inches and below have cast-iron flanges. Above 20 inches have forged steel ring flanges.

Flanges recessed so that copper projects 1/16 inch beyond face of flange. Copper lap carried out on flanges to inside edge of bolt hole.

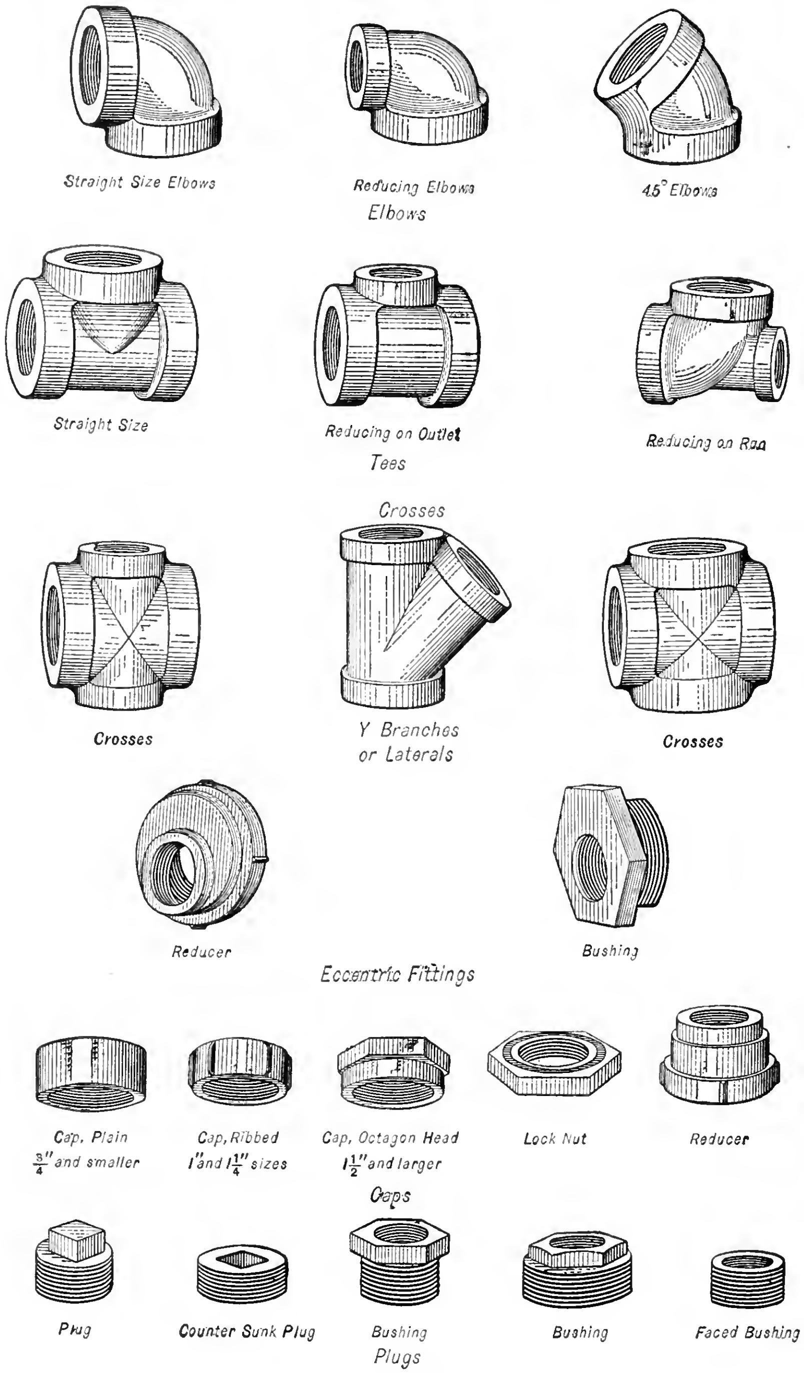


FIG. 6. CAST-IRON FITTINGS.

FITTINGS

Commercial Classifications of Fittings. Commercial fittings for joining the separate lengths of pipe together are made in a great variety of forms, and are either *screwed* or *flanged;* the former being generally used for the smaller sizes of pipe up to and including 3½″, and the latter for

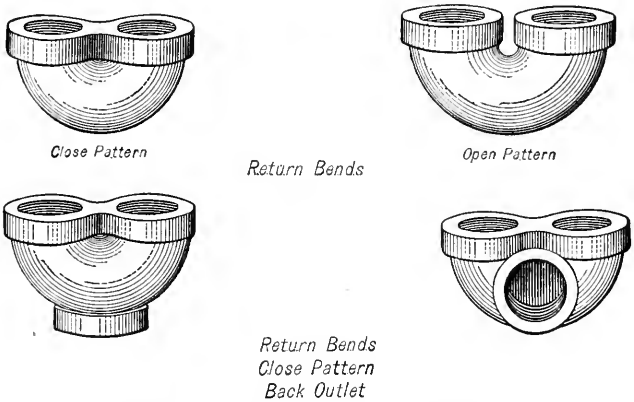


FIG. 6a. CAST-IRON FITTINGS. (*Continued*)

the larger sizes, 4″ and above. Screwed fittings of large size as well as flanged fittings of small size are also made, however, and used for certain classes of work at the proper pressure.

The material used for fittings is generally *cast iron*, but in addition to this *malleable iron*, *steel* and *steel alloys* are also used, as well as various grades of *brass*. The material to be used depends on the character of the service and the pressure. In this connection see the Specifications in the chapter on "Power Plant Piping."*

As in the case of pipe there are several weights of fittings manufactured designed to be used with pipe of corresponding grade. These variations in weights are known as (1) *low-pressure* fittings for steam working pressures of 25 lb., (2) *standard* fittings for steam working pressures of 125 lb., and *extra heavy* fittings for steam working pressures of 250 lb. These latter fittings are suitable for water working pressures of 350 lb., and are usually tested to twice the steam working pressure or 500 lb. cold hydrostatic.

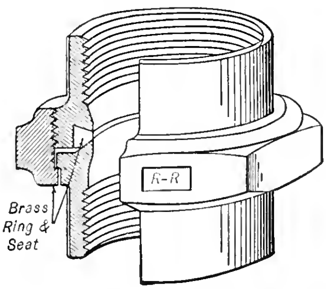


FIG. 7. COMBINATION SCREWED UNION.

Screwed Fittings. Screwed fittings include *nipples* or short pieces of pipe of varying lengths; *couplings*, usually of wrought iron only; *elbows* for turning angles of either 45° or 90°; *return bends*, which may be of either the *close* or *open* pattern, and may be cast with either a *back* or *side outlet; tees; crosses; laterals* or **Y** *branches;* and a variety of *plugs, bushings, caps, lock-nuts, flanges* and *reducing fittings*, as shown in Fig. 6. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped *eccentrically* to permit of free drainage of the water of condensation in steam lines or free escape of air in water lines.

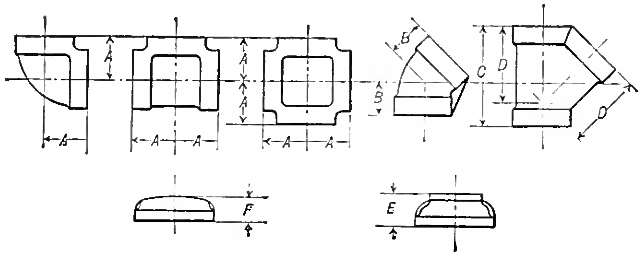
In many lines it is necessary to make provision for disconnection, and a special fitting called a *union* (Fig. 7), of which there are many modifications, is used in this case. This fitting serves

*See Volume II.

TABLE 7

GENERAL DIMENSIONS OF STANDARD CAST-IRON SCREWED FITTINGS

(Crane Co.)



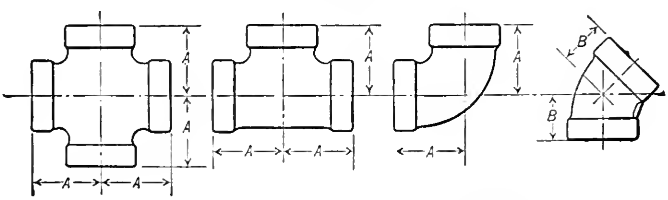
Size, Inches	Dimensions *A* Inches	Dimensions *B* Inches	Dimensions *C* Inches	Dimensions *D* Inches	Dimensions *E* Inches	Dimensions *F* Inches
1/4	13/16	3/4	....	....	...	...
3/8	15/16	13/16	....	....	...	...
1/2	1 1/8	7/8	2 1/2	1 7/8	...	...
3/4	1 5/16	1	3	2 1/4	...	...
1	1 7/16	1 1/8	3 1/2	2 3/4	...	...
1 1/4	1 3/4	1 5/16	4 1/4	3 1/4	...	...
1 1/2	1 15/16	1 7/16	4 7/8	3 13/16	...	...
2	2 1/4	1 11/16	5 3/4	4 1/2	...	...
2 1/2	2 11/16	1 15/16	6 1/4	5 3/16	...	...
3	3 1/8	2 3/16	7 7/8	6 1/8	2 15/16	...
3 1/2	3 7/16	2 3/8	8 7/8	6 7/8	3 1/8	...
4	3 3/4	2 5/8	9 3/4	7 5/8	3 3/8	2 1/16
4 1/2	4 1/16	2 13/16	11 5/8	9 1/4	3 5/8	2 3/16
5	4 7/16	3 1/16	11 5/8	9 1/4	3 7/8	2 3/8
6	5 1/8	3 7/16	13 7/16	10 3/4	4 3/8	2 5/8
7	5 13/16	3 7/8	15 1/4	12 1/4	4 13/16	2 7/8
8	6 1/2	4 1/4	16 15/16	13 5/8	5 1/4	3 1/8
9	7 3/16	4 11/16	20 11/16	16 3/4	5 11/16	3 3/8
10	7 7/8	5 3/16	20 11/16	16 3/4	6 3/16	3 5/8
12	9 1/4	6	24 1/8	19 5/8	7 1/8	4 1/4

Note.—The above dimensions are subject to a slight variation.

TABLE 8

GENERAL DIMENSIONS OF EXTRA HEAVY CAST-IRON SCREWED FITTINGS

(Crane Co.)



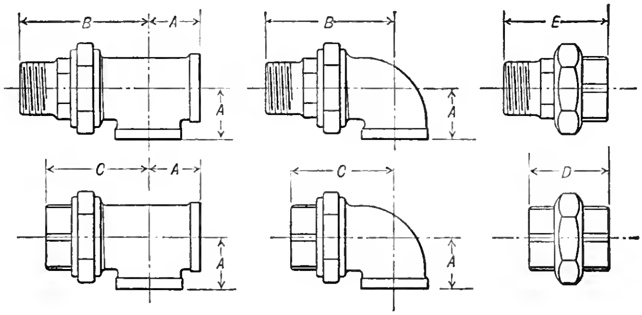
Size, Inches	Dimensions *A* Inches	Dimensions *B* Inches
1	2	1 3/8
1 1/4	2 1/4	1 1/2
1 1/2	2 9/16	1 5/8
2	3	1 15/16
2 1/2	3 1/2	2 1/4
3	4 1/8	2 1/2
3 1/2	4 11/16	2 9/16
4	5 1/8	2 3/4
4 1/2	5 1/2	3
5	6 1/8	3 5/16
6	7 1/4	3 3/4
7	8 1/8	4
8	9 1/8	4 3/4
10	11 3/8	4 7/8
12	13 3/8	5 1/2

Note.—The above dimensions are subject to a slight variation.

TABLE 9

GENERAL DIMENSIONS OF MALLEABLE IRON UNIONS, UNION ELBOWS AND UNION TEES

(Crane Co.)



Size, Inches	Center to End			End to End			
	Dimensions *A*	Dimensions *B*	Dimensions *C*	Dimensions *D*	Dimensions *D*	Dimensions *D*	Dimensions *E*
	Elbows and Tees, Inches	Union, Male, Inches	Union, Female, Inches	Standard Union, Inches	Railroad and Chicago Unions, Inches	Crane and Navy Unions, Inches	Standard and Railroad Unions with Male and Female Ends, Inches
1/8	...	...	...	...	1 1/2	...	...
1/4	13/16	2 7/16	1 13/16	1 5/8	1 5/8	2 3/16	2 1/4
3/8	15/16	2 3/4	2 1/16	1 3/4	1 3/4	2 1/4	2 7/16
1/2	1 1/8	3 1/16	2 5/16	1 7/8	1 7/8	2 5/16	2 11/16
3/4	1 5/16	3 1/2	2 5/8	2 1/8	2 1/8	2 1/2	3 1/16
1	1 7/16	3 15/16	3	2 3/8	2 3/8	2 5/8	3 5/16
1 1/4	1 3/4	4 7/16	3 7/16	2 5/8	2 5/8	2 29/32	3 11/16
1 1/2	1 15/16	4 3/4	3 13/16	2 15/16	2 15/16	3 1/8	4
2	2 1/4	5 3/8	4 3/8	3 1/4	3 1/4	3 9/16	4 5/16
2 1/2	2 13/16	6	4 7/8	3 9/16	3 9/16	4 1/16	4 5/8
3	...	...	...	3 15/16	3 15/16	4 7/8	...
3 1/2	...	...	...	4 3/8	...	...	...
4	...	...	...	4 1/2	...	...	...

the same purpose as a pair of bolted flanges and is seldom made or used in sizes above 4″. Flanges are used generally on lines 3″ in diameter and larger. Unions are usually designed with a brass seat so that the sleeves will not rust fast together.

The union may be applied to and made a part of other fittings and valves in order to facilitate their disconnection. Combination fittings such as *union tees and union elbows* are shown and dimensions given in Table 9.

All fittings are *threaded* to conform with standard pipe threads, using *Briggs* standard gage and taper, and, unless otherwise specified, *right-hand threads* are used. Fittings with *left-hand* or *right- and left-hand* threads usually have some distinguishing mark cast upon them, and must be so specified.

In addition to the ordinary *close fittings* shown in Fig. 6 it is possible to obtain *long-sweep* or *long-turn* fittings designed to materially reduce the friction loss occasioned by close fittings. These fittings are made only in the standard weight.

Malleable-iron fittings, like *brass fittings,* are cast with a round instead of a flat band or bead, or with no bead at all, that is, perfectly plain. They are likely to be less porous than cast-iron fittings.

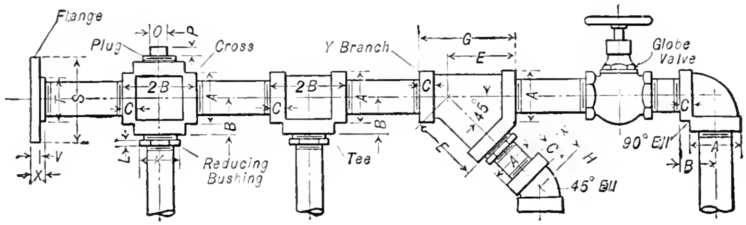
Fittings are designated as *male* or *female*, depending on whether the threads are on the outside or the inside, as shown in Table 9, where the male fittings are shown at the top.

Space Required by Screwed Fittings. The space required for fittings and branch connections, and the application of these fittings to an actual pipe line is shown in the figures at the head of Tables 10 and 11. The minimum clearance dimensions are also given in Tables 10 and 11, and will be found useful in laying out steam and water piping where screwed fittings are to be employed.

TABLE 10

APPLICATIONS OF CAST-IRON SCREWED FITTINGS

(Dimensions in inches)



Size of Pipe	A	B	C	E	G	H	K	L	O	P	S	T	V	X
1/8	31/32	3/4	5/16	1 3/16	1 13/16	21/32	...	..	1/4	1/4	....	...	...	...
1/4	1 1/8	29/32	13/32	1 7/16	2 7/32	25/32	...	..	3/8	1/4	....	...	...	...
3/8	1 3/8	1 1/16	1/2	1 11/16	2 19/32	29/32	13/16	1/4	3/8	5/16	....	...	...	...
1/2	1 5/8	1 1/4	9/16	2 1/32	3 1/16	1 1/32	1 3/32	1/4	3/8	5/16	....	...	...	...
3/4	1 15/16	1 7/16	21/32	2 13/32	3 21/32	1 3/16	1 9/32	1/4	7/16	3/8	3 1/2	1 1/2	7/16	5/8
1	2 5/16	1 11/16	3/4	2 29/32	4 9/32	1 3/8	1 21/32	3/8	11/16	7/16	4	1 15/16	7/16	11/16
1 1/4	2 3/4	2	27/32	3 1/2	5 3/32	1 19/32	2	3/8	3/4	1/2	4 1/2	2 5/16	1/2	3/4
1 1/2	3 1/8	2 3/16	20/32	3 29/32	5 19/32	1 11/16	2 5/32	3/8	13/16	1/2	5	2 5/8	9/16	7/8
2	3 11/16	2 19/32	1 1/32	4 3/4	6 3/4	2	2 3/4	3/8	1 1/16	9/16	6	3 1/8	5/8	1
2 1/2	4 5/16	2 31/32	1 5/32	5 9/16	7 13/16	2 1/4	3 15/32	7/16	1 3/15	5/8	7	3 5/8	11/16	1 1/16
3	5 1/32	3 11/32	1 1/4	6 15/32	8 15/16	2 15/32	4 1/32	1/2	1 5/16	5/8	7 1/2	4 5/16	3/4	1 1/8
3 1/2	5 23/32	3 3/4	1 3/8	7 5/16	10 1/16	2 3/4	5 3/16	1/2	1 1/2	3/4	8 1/2	4 7/8	13/16	1 3/16
4	6 3/8	4 1/8	1 1/2	8 5/16	11 5/16	3	5 29/32	1/2	1 3/4	3/4	9	5 3/8	15/16	1 1/4
4 1/2	7 1/32	4 1/2	1 19/32	8 15/16	12 3/16	3 1/4	6 1/32	9/16	2	13/16	9 1/4	5 13/16	15/16	1 5/16
5	7 3/4	4 27/32	1 11/16	9 3/4	12 5/8	3 15/32	7 3/16	9/16	2 1/4	13/16	10	6 7/16	15/16	1 7/16
6	9	5 17/32	1 13/32	11 5/16	15 7/32	3 29/32	8 5/8	3/4	2 1/2	1	11	7 9/16	1	1 1/2

Method of Designating Reducing Fittings. In describing reducing fittings with more than two threaded openings for pipe connections, it is very necessary to specify these openings in a certain recognized order. For example, a reducing tee is always denoted by the run, first beginning with the larger opening, and ending with the side outlet, as in Fig. 8, where the tee shown is a 10″ x 8″ x 6″. In the case of a cross, read through one way, or the run, and then through the other way, as in Fig. 9, where the cross is a 10″ x 8″ x 6″ x 5″.

Tees in which the side outlet is larger than the openings through the run are known as *bull-head tees,* and are specified in the same manner as for reducing tees.

Reducing flanges are always specified by naming the smaller pipe size first, and then the outside diameter of the flange, as a 10″ x 19″ flange, which means that a 10″ line is to be connected to a 12″ valve or fitting since a 12″ standard flange is 19″ across its outside diameter. Do not use the larger pipe size at all.

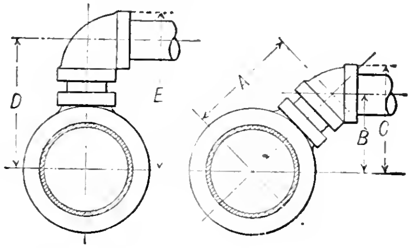
Flanged Fittings. Flanged fittings are generally used in the best practice for connecting all piping above 6″ in diameter. While screwed fittings may be used for the larger sizes, and

TABLE 11

SPACE REQUIRED FOR BRANCH CONNECTIONS

Minimum Height of Connections off Pipe Mains

Dimensions given in inches.



Mains	Branches	*A*	*B*	*C*	*D*	*E*	Branches	Mains
2	1	3 3/8	2 3/8	3 13/32	3 31/32	5	1	2
2	1 1/4	3 11/16	2 5/8	3 7/8	4 7/16	5 11/16	1 1/4	2
2	1 1/2	4	2 27/32	4 7/32	4 13/16	6 3/16	1 1/2	2
2 1/2	1	3 3/4	2 21/32	3 11/16	4 11/32	5 3/8	1	2 1/2
2 1/2	1 1/4	4 1/16	2 7/8	4 1/8	4 13/16	6 1/16	1 1/4	2 1/2
2 1/2	1 1/2	4 3/8	3 3/32	4 15/32	5 3/16	6 9/16	1 1/2	2 1/2
2 1/2	2	4 7/8	3 7/16	5 1/8	5 7/8	7 9/16	2	2 1/2
3	1	4 1/16	2 7/8	3 29/32	4 21/32	5 11/16	1	3
3	1 1/4	4 3/8	3 3/32	4 11/32	5 1/8	6 3/8	1 1/4	3
3	1 1/2	4 11/16	3 5/16	4 11/16	5 1/2	6 7/8	1 1/2	3
3	2	5 3/16	3 11/16	5 3/8	6 3/16	7 7/8	2	3
3	2 1/2	5 9/16	3 15/16	6	6 13/16	8 7/8	2 1/2	3
3 1/2	1	4 11/32	3 13/16	4 3/32	4 15/16	5 31/32	1	3 1/2
3 1/2	1 1/4	4 21/32	3 5/16	4 9/16	5 13/32	6 21/32	1 1/4	3 1/2
3 1/2	1 1/2	4 31/32	3 17/32	4 29/32	5 25/32	7 5/32	1 1/2	3 1/2
3 1/2	2	5 15/32	3 7/8	5 9/16	6 15/32	8 5/32	2	3 1/2
3 1/2	2 1/2	5 27/32	4 1/8	6 3/16	7 3/32	9 5/32	2 1/2	3 1/2
4	1	4 11/16	3 5/16	4 11/32	5 9/32	6 5/16	1	4
4	1 1/4	5	3 17/32	4 25/32	5 3/4	7	1 1/4	4
4	1 1/2	5 5/16	3 3/4	5 1/8	6 1/8	7 1/2	1 1/2	4
4	2	5 13/16	4 1/8	5 13/16	6 13/16	8 1/2	2	4
4	2 1/2	6 3/16	4 3/8	6 7/16	7 7/16	9 1/2	2 1/2	4
5	1 1/4	5 17/32	3 29/32	5 5/32	6 9/32	7 17/32	1 1/4	5
5	1 1/2	5 27/32	4 1/8	5 1/2	6 21/32	8 1/32	1 1/2	5
5	2	6 11/32	4 1/2	6 3/16	7 11/32	9 1/32	2	5
5	2 1/2	6 23/32	4 3/4	6 13/16	7 31/32	10 1/32	2 1/2	5
6	1 1/4	6 3/16	4 3/8	5 5/8	6 15/16	8 3/16	1 1/4	6
6	1 1/2	6 1/2	4 5/8	6	7 5/16	8 11/16	1 1/2	6
6	2	7	4 31/32	6 21/32	8	9 11/16	2	6
6	2 1/2	7 3/8	5 7/32	7 9/32	8 5/8	10 11/16	2 1/2	6
8	2	8 1/4	5 27/32	7 17/32	9 1/4	10 15/16	2	8
8	2 1/2	8 5/8	6 1/8	8 3/16	9 7/8	11 15/16	2 1/2	8
8	3	9	6 3/8	8 3/4	10 7/16	12 13/16	3	8

NOTE.—Table prepared by *Fred'k D. B. Ingalls, M.E.*

are perfectly satisfactory under the proper working conditions, it will be found difficult to either make or break the joints in these large sizes.

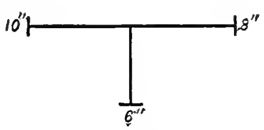


FIG. 8. TEE 10" x 8" x 6".

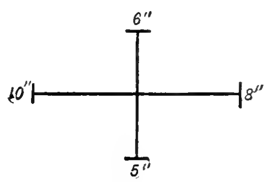


FIG. 9. CROSS 10" x 8" x 6" x 5".

In order to secure uniformity in dimensions, weight, and bolting of flanged fittings, attempts have been made to secure uniform standards, with only partial success, until Jan. 1, 1914, when

the *American Standard* went into effect with the sanction of the *A. S. M. E.* and the principal manufacturers of flanged fittings.

This new schedule provides for two grades of fittings, to be known as *standard* and *extra heavy*, suitable for steam working pressures up to 125 and 250 lb. respectively. The following explanatory notes, together with the dimension tables, give all necessary data relating to these fittings in sizes from 1″ to 48″ diameter.

Notes on the American Standard for Flanged Fittings. (See Tables of Dimensions.)

1. Standard and extra heavy reducing elbows carry same dimensions center to face as regular elbows of largest straight size.

2. Standard and extra heavy tees, crosses and laterals, reducing on run *only*, carry same dimensions face to face as largest straight size.

3. If flanged fittings for lower working pressure than 125 lb. are made, they shall conform in all dimensions, except thickness of shell, to this standard and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be standard dimensions.

4. Where long radius fittings are specified, it has reference only to elbows which are made in two center to face dimensions and to be known as elbows and long radius elbows, the latter being used only when so specified.

5. All standard weight fittings must be guaranteed for 125 lb. working pressure and extra heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and guaranteed working steam pressure.

6. All extra heavy fittings and flanges to have a raised surface of $1/16$″ high inside of bolt holes for gaskets.

Standard weight fittings and flanges to be plain faced.

Bolt holes to be $\frac{1}{8}$″ larger in diameter than bolts.

Bolt holes to straddle center line.

7. Size of all fittings scheduled indicates inside diameter of ports, except for extra heavy fitting 14″ and larger, when the port diameter is $\frac{3}{4}$″ smaller than nominal size.

8. The face to face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in tables of dimensions.

9. Square head bolts with hexagonal nuts are recommended.

For bolts $1\frac{5}{8}$″ diameter and larger, studs with a nut on each end are satisfactory.

Hexagonal nuts for pipe sizes 1″ to 46″ on 125 lb. standard, and 1″ to 16″ on 250 lb. standard, can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48″ to 100″ on 125 lb., and 18″ to 48″ on 250 lb., pound standard, can be conveniently pulled up with box wrenches.

10. Twin elbows, whether straight or reducing, carry same dimensions center to face and face to face as regular straight size ells and tees.

Side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tees having same reductions.

11. Bull head tees or tees increasing on outlet will have same center to face and face to face dimensions as a straight fitting of the size of the outlet.

12. Tees and crosses 9″ and down, reducing on the outlet, use the same dimensions as straight sizes of the larger port.

Sizes 10″ and up, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.

Laterals $3\frac{1}{2}$″ and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.

13. Sizes 4″ and up, reducing on the branch, are made in two lengths, depending on the size of the branch as given in the tables of dimensions.

The dimensions of reducing flanged fittings are always regulated by the reductions of the outlet or branch. Fittings reducing on the run only, the long body pattern will always be used.

Y's are special and are made to suit conditions.

Double sweep tees are not made reducing on the run.

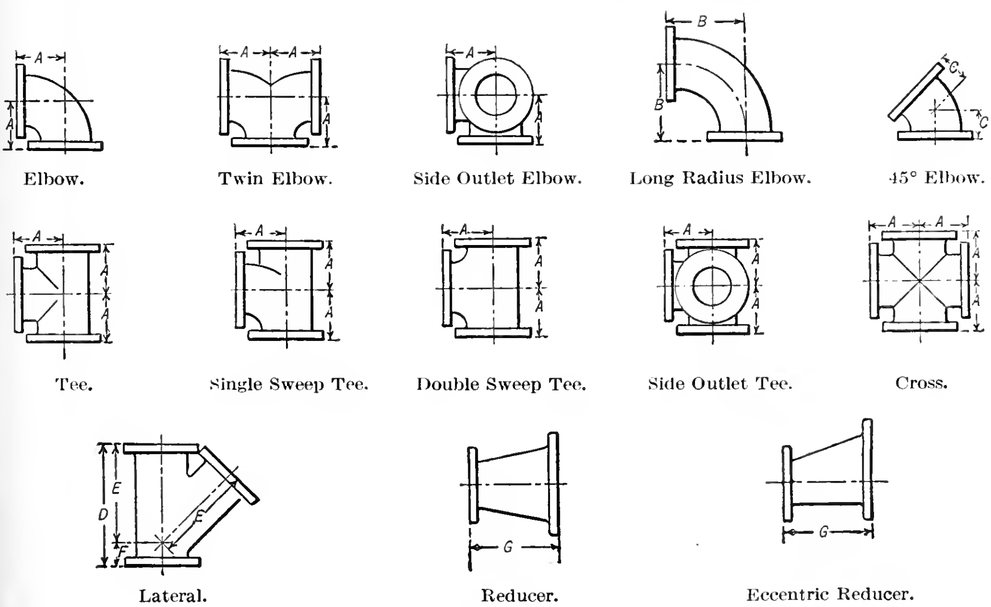
14. *Steel flanges, fittings, and valves are recommended for superheated steam.*

TABLE 12

AMERICAN STANDARD
STANDARD FLANGED FITTINGS
General Dimensions—Straight Sizes
All dimensions given in inches
(See Fig. 10)

Size	1	1 1/4	1 1/2	2	2 1/2	3	3 1/2	4	4 1/2	5	6	7	8	9	10	12	14	15
AA—Face to Face—Tees and Crosses	7	7 1/2	8	9	10	11	12	13	14	15	16	17	18	20	22	24	28	29
A—Center to Face—Ells, Tees and Crosses	3 1/2	3 3/4	4	4 1/2	5	5 1/2	6	6 1/2	7	7 1/2	8	8 1/2	9	10	11	12	14	14 1/2
B—Center to Face—Long Radius Ells	5	5 1/2	6	6 1/2	7	7 3/4	8 1/2	9	9 1/2	10 1/4	11 1/2	12 3/4	14	15 1/4	16 1/2	19	21 1/2	22 3/4
C—Center to Face—45° Ells	1 3/4	2	2 1/4	2 1/2	3	3	3 1/2	4	4	4 1/2	5	5 1/2	5 1/2	6	6 1/2	7 1/2	7 1/2	8
D—Face to Face—Laterals	7 1/2	8	9	10 1/2	12	13	14 1/2	15	15 1/2	17	18	20 1/2	22	24	25 1/2	30	33	34 1/2
E—Center to Face—Laterals	5 3/4	6 1/4	7	8	9 1/2	10	11 1/2	12	12 1/2	13 1/2	14 1/2	16 1/2	17 1/2	19 1/2	20 1/2	24 1/2	27	28 1/2
F—Center to Face—Laterals	1 3/4	1 3/4	2	2 1/2	2 1/2	3	3	3	3	3 1/2	3 1/2	4	4 1/2	4 1/2	5	5 1/2	6	6
G—Face to Face—Reducers	....	....	....	....	....	6	6 1/2	7	7 1/2	8	9	10	11	11 1/2	12	14	16	17
Diameter of Flanges	4	4 1/2	5	6	7	7 1/2	8 1/2	9	9 1/4	10	11	12 1/2	13 1/2	15	16	19	21	22 1/4
Thickness of Flanges	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	15/16	1	1 1/16	1 1/8	1 1/8	1 3/16	1 1/4	1 3/8	1 3/8
Minimum Metal Thickness of Body	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	1/2	1/2	9/16	5/8	5/8	11/16	3/4	13/16	7/8	7/8

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
AA—Face to Face—Tees and Crosses	30	33	36	40	44	46	48	50	52	54	56	58	60	62	64	66	68
A—Center to Face—Ells, Tees and Crosses	15	16 1/2	18	20	22	23	24	25	26	27	28	29	30	31	32	33	34
B—Center to Face—Long Radius Ells	24	26 1/2	29	31 1/2	34	36 1/2	39	41 1/2	44	46 1/2	49	51 1/2	54	56 1/2	59	61 1/2	64
C—Center to Face—45° Ells	8	8 1/2	9 1/2	10	11	13	14	15	16	17	18	19	20	21	22	23	24
D—Face to Face—Laterals	36 1/2	39	43	46	49 1/2	53	56	59	....	....	....	....	....	....	....	....	....
E—Center to Face—Laterals	30	32	35	37 1/2	40 1/2	44	46 1/2	49	....	....	....	....	....	....	....	....	....
F—Center to Face—Laterals	6 1/2	7	8	8 1/2	9	9	9 1/2	10	....	....	....	....	....	....	....	....	....
G—Face to Face—Reducers	18	19	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
Diameter of Flanges	23 1/2	25	27 1/2	29 1/2	32	34 1/4	36 1/2	38 3/4	41 3/4	43 3/4	46	48 3/4	50 3/4	53	55 1/4	57 1/4	59 1/2
Thickness of Flanges	1 7/16	1 9/16	1 11/16	1 13/16	1 7/8	2	2 1/16	2 1/8	2 1/4	2 5/16	2 3/8	2 3/8	2 1/2	2 5/8	2 5/8	2 11/16	2 3/4
Minimum Metal Thickness of Body	1	1 1/16	1 1/8	1 3/16	1 1/4	1 5/16	1 3/8	1 7/16	1 1/2	1 9/16	1 5/8	1 11/16	1 3/4	1 13/16	1 7/8	1 15/16	2



Elbow. Twin Elbow. Side Outlet Elbow. Long Radius Elbow. 45° Elbow.

Tee. Single Sweep Tee. Double Sweep Tee. Side Outlet Tee. Cross.

Lateral. Reducer. Eccentric Reducer.

FIG. 10. STANDARD AND EXTRA HEAVY FLANGED FITTINGS (*American Standard.*)

TABLE 13

EXTRA HEAVY FLANGED FITTINGS

General Dimensions—Straight Sizes

All dimensions given in inches

(See Fig. 10)

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
AA—Face to Face—Tees and Crosses	8	8½	9	10	11	12	13	14	15	16	17	18	20	21	....	....	30	31
A—Center to Face—Ells, Tees and Crosses	4	4¼	4½	5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	....	15	15½
B—Center to Face—Long Radius Ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾
C—Center to Face—45° Ells	2	2½	2¾	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9
D—Face to Face—Laterals	8½	9½	11	11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½	39½
E—Center to Face—Laterals	6½	7¼	8½	9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31	33
F—Center to Face—Laterals	2	2¼	2½	2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½	6½
G—Face to Face—Reducers	....	....	....	....	....	6	6½	7	7½	8	9	10	11	11½	12	14	16	17
Diameter of Flanges	4½	5	6	6½	7½	8¼	9	10	10½	11	12½	14	15	16¼	17½	20½	23	24½
Thickness of Flanges	11/16	¾	13/16	⅞	1	1⅛	1 3/16	1¼	1 5/16	1⅜	1 7/16	1½	1⅝	1¾	1⅞	2	2⅛	2 3/16
Minimum Metal Thickness of Body	½	½	½	½	9/16	9/16	9/16	⅝	⅝	11/16	¾	13/16	13/16	⅞	15/16	1	1⅛	1 3/16

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
AA—Face to Face—Tees and Crosses	33	36	39	41	45	48	52	55	58	61	65	68	71	74	78	81	84
A—Center to Face—Ells, Tees and Crosses	16½	18	19½	20½	22½	24	26	27½	29	30½	32½	34	35½	37	39	40½	42
B—Center to Face—Long Radius Ells	24	26½	29	31½	34	36½	39	41½	44	46½	49	51½	54	56½	59	61½	64
C—Center to Face—45° Ells	9½	10	10½	11	12	13	14	15	16	17	18	19	20	21	22	23	24
D—Face to Face—Laterals	42	45½	49	53	57½	....	....	....	....	....	....	....	....	....	....	....	....
E—Center to Face—Laterals	34½	37½	40½	43½	47½	....	....	....	....	....	....	....	....	....	....	....	....
F—Center to Face—Laterals	7½	8	8½	9½	10	....	....	....	....	....	....	....	....	....	....	....	....
G—Face to Face—Reducers	18	19	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
Diameter of Flanges	25½	28	30½	33	36	38¼	40¾	43	45¼	47½	50	52¼	54½	57	59¼	61½	65
Thickness of Flanges	2¼	2⅜	2½	2⅝	2¾	2 13/16	2 15/16	3	3⅛	3¼	3⅜	3 7/16	3 9/16	3 11/16	3¾	3⅞	4
Minimum Metal Thickness of Body	1¼	1⅜	1½	1 9/16	1⅝	1 13/16	1⅞	2	2⅛	2¼	2⅜	2 7/16	2 9/16	2 11/16	2 13/16	2⅞	3

TABLE 14

AMERICAN STANDARD

STANDARD REDUCING FLANGE FITTINGS

General Dimensions—Reducing Tees and Crosses

All dimensions given in inches

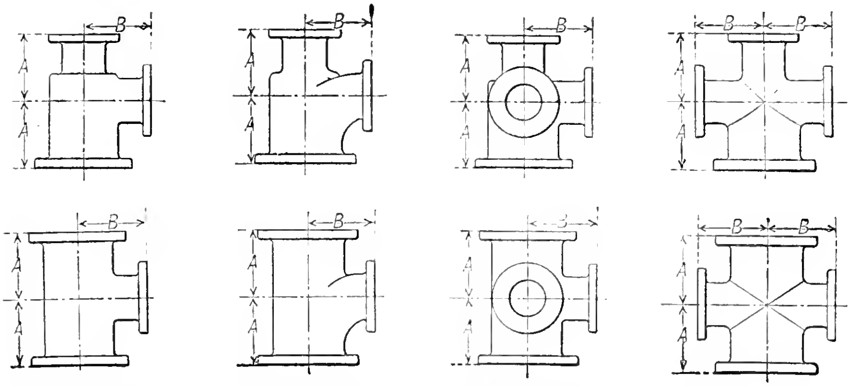
(See Fig. 11)

Short Body Pattern

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
* Size of Outlets and Smaller	All reducing fittings 1″–9″ inclusive have the same center to face dimensions as straight size fittings														6	8	9	9
AA—Face to Face, Run															18	20	22	23
A—Center to Face, Run															9	10	11	11½
B—Center to Face, Outlet															9½	11	13	13½

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
* Size of Outlets and Smaller	10	12	14	15	16	18	18	20	20	22	24	24	26	28	28	30	32
AA—Face to Face, Run	24	26	28	28	30	32	32	36	36	38	40	40	44	46	46	48	52
A—Center to Face, Run	12	13	14	14	15	16	16	18	18	19	20	20	22	23	23	24	26
B—Center to Face, Outlet	14	15½	17	18	19	20	21	23	24	25	26	28	29	30	31	33	34

* Long body patterns are used when outlets are larger than given in the above table, therefore have same dimensions as straight size fittings.



Reducing Tees. Reducing Single Sweep Tees. Reducing Side Outlet Tees. Reducing Crosses.

FIG. 11. STANDARD AND EXTRA HEAVY REDUCING FLANGED FITTINGS

TABLE 15

EXTRA HEAVY REDUCING FLANGED FITTINGS

(See Fig. 11)

Short Body Pattern

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
* Size of Outlets and Smaller	All reducing fittings 1″–9″ inclusive have the same center to face dimensions as straight size fittings.														6	8	9	9
AA—Face to Face, Run															18	21	23	23
A—Center to Face, Run															9	10½	11½	11½
B—Center to Face, Outlet															11	12½	14	15

Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
* Size of Outlets and Smaller	10	12	14	15	16	18	18	20	20	22	24	24	26	28	28	30	32
AA—Face to Face, Run	25	28	31	33	34	38	38	41	41	44	47	47	50	53	53	55	58
A—Center to Face, Run	12½	14	15½	16½	17	19	19	20½	20½	22	23½	23½	25	26½	26½	27½	29
B—Center to Face, Outlet	15½	17	18½	20	21½	23	24	25½	26½	28	29½	30½	31½	33½	34½	35½	37½

* Long body patterns are used when outlets are larger than given in the above table, therefore have same dimensions as straight size fittings.

The dimensions of Reducing Flanged Fittings are always regulated by the reduction of the outlet.

Fittings Reducing on the Run Only, the long body pattern will always be used, except Double Sweep Tees, on which the reduced end (dimension on request) is always longer than the regular fitting.

Bull Heads or Tees, having outlet larger than the run, will be the same length center to face of all openings as a Tee with all openings of the size of the outlet.

For example, a 12″ x 12″ x 18″ Tee will be governed by the dimensions of the 18″ Long Body Tee; namely, 16½″ center to face of all openings and 33″ face to face.

Reducing Elbows carry same center to face dimensions as regular elbows of largest straight size.

Special Flanged Fittings. In addition to the standard flanged fittings it is often desirable to make use of *special cast-iron flanged fittings* (Fig. 13) to fit peculiar conditions, and save an unnecessary amount of piping. Such fittings are not carried in stock but will be made to order according to detail drawing, giving dimensions called for in outline figures.

TABLE 16

AMERICAN STANDARD

STANDARD FLANGED FITTINGS

General Dimensions—Reducing Laterals. (See Fig. 12.) All Dimensions given in inches

FIG. 12.

Short Body Pattern

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24	26	28	30
*Size of Branches & Smaller	All reducing fittings 1"–3½" inclusive have the same center to face dimensions as straight size fittings.							2½	2½	3	3	3½	4	4½	5	6	7	7	8	9	10	10	12	12	14	15
C—Face to Face, Run								13	13	14	15	16	16	17	18	20	22	23	24	26	28	29	32	35	37	39
D—Center to Face, Run								11	11	12	13½	14½	14½	15½	17	19	21	22	23	25	27	28½	31½	35	37	39
E—Center to Face, Run								2	2	2	1½	1½	1½	1½	1	1	1	1	1	1	1	½	½	0	0	0
F—Center Face, Branch								11	11	12	13½	15	15½	16½	18	20½	23	24	25½	27½	29½	31½	34½	38	40	42

TABLE 17

EXTRA HEAVY FLANGED FITTINGS

General Dimensions—Reducing Laterals. (See Fig. 12.) All Dimensions given in inches

Short Body Pattern

Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24
*Size of Branch and Smaller	All reducing fittings 1" to 3½" inclusive have the same center to face dimensions as straight size fittings.							2½	2½	3	3	3½	4	4½	5	6	7	7	8	9	10	10	12
C—Face to Face, Run								14	15	16	17	18	20	21	23	26	29	30	32	34	37	40	44
D—Center to Face, Run								12	12½	13½	14½	15½	17½	18½	20½	23½	26½	27½	29	31	34	37	41
E—Center to Face, Run								2	2½	2½	2½	2½	2½	2½	2½	2½	2½	2½	3	3	3	3	3
F—Center to Face, Branch								13	13½	14½	15½	16½	18½	19½	21½	24½	27½	28½	30½	32½	36	39	43

***Long body patterns are used when branches are larger than given in the above table, therefore have same dimensions as straight size fittings.**

The dimensions of Reducing Flanged Fittings are always regulated by the reduction of the branch.

Fittings reducing on the run only, the long body pattern will always be used.

TABLE 18

AMERICAN STANDARD

FLANGED VALVES AND FITTINGS—STANDARD AND LOW PRESSURE

Templates for Drilling

Size, Inches	Diameter of Flanges, Inches	Thickness of Flanges, Inches	Bolt Circle, Inches	Number of Bolts	Size of Bolts, Inches
1	4	7/16	3	4	7/16
1¼	4½	½	3⅜	4	7/16
1½	5	9/16	3⅞	4	½
2	6	⅝	4¾	4	⅝
2½	7	11/16	5½	4	⅝
3	7½	¾	6	4	⅝
3½	8½	13/16	7	4	⅝
4	9	15/16	7½	8	⅝
4½	9¼	15/16	7¾	8	¾
5	10	15/16	8½	8	¾
6	11	1	9½	8	¾
7	12½	1 1/16	10¾	8	¾
8	13½	1⅛	11¾	8	¾
9	15	1⅛	13¼	12	¾
10	16	1 3/16	14¼	12	⅞
12	19	1¼	17	12	⅞
14	21	1⅜	18¾	12	1
15	22¼	1⅜	20	16	1
16	23½	1 7/16	21¼	16	1
18	25	1 9/16	22¾	16	1⅛
20	27½	1 11/16	25	20	1⅛
22	29½	1 13/16	27¼	20	1¼
24	32	1⅞	29½	20	1¼
26	34¼	2	31¾	24	1¼
28	36½	2 1/16	34	28	1¼
30	38¾	2⅛	36	28	1⅜
32	41¾	2¼	38½	28	1½
34	43¾	2 5/16	40½	32	1½
36	46	2⅜	42¾	32	1½
38	48¾	2⅜	45¼	32	1⅝
40	50¾	2½	47¼	36	1⅝
42	53	2⅝	49½	36	1⅝
44	55¼	2⅝	51¾	40	1⅝
46	57¼	2 11/16	53¾	40	1⅝
48	59½	2¾	56	44	1⅝

Flanges for Wrought Pipe. Flanges of cast iron and steel are used for connecting pipes of large diameter, and the principal dimensions, including bolt circle and bolts, are given in Tables 18 and 19. There are in general four standard methods of joining these flanges to the pipe, as shown in Fig. 14, and they are known as: (1) screwed flanges, (2) shrunk flanges, or shrunk and rolled flanges, (3) Vanstone joints, plain or reinforced, and (4) welded flanges.

Screwed Flanges. A method of attaching a flange, provided with threads to the end of a pipe provided with threads, by screwing on the flange by machinery until the pipe extends beyond the face of the flange, then swinging pipe in a double-ended lathe and facing off both ends of pipe and at the same time taking a skim off the face of the flanges to insure a true bearing of gasket on end of pipe and parallelism of flanges is shown by *A*, Fig. 14.

This joint is adaptable for medium steam pressures and high water pressures in sizes up to 12″. Manufacturers do not recommend this type of flange in sizes 14″ and larger for high pressures, as practice has proven it to be both unsatisfactory and unsafe.

Shrunk and Rolled Flanges. For pipe sizes 14″ and larger, and for pipe where the thickness of wall of pipe was too light for successful threading, the only practical method of attaching flanges was by riveting or shrinking and peening, as there were no threading facilities capable of taking care of the larger sizes, for many years, except by chasing the threads, which was too expensive for ordinary use.

TABLE 19

AMERICAN STANDARD

FLANGED VALVES AND FITTINGS—EXTRA HEAVY

Templates for Drilling

Size, Inches	Diameter of Flanges, Inches	Thickness of Flanges, Inches	Bolt Circle Inches	Number of Bolts	Size of Bolts, Inches
1	4½	11/16	3¼	4	½
1¼	5	¾	3¾	4	½
1½	6	13/16	4½	4	⅝
2	6½	⅞	5	4	⅝
2½	7½	1	5⅞	4	¾
3	8¼	1⅛	6⅝	8	¾
3½	9	1 3/16	7¼	8	¾
4	10	1¼	7⅞	8	¾
4½	10½	1 5/16	8½	8	¾
5	11	1⅜	9¼	8	¾
6	12½	1 7/16	10⅝	12	¾
7	14	1½	11⅞	12	⅞
8	15	1⅝	13	12	⅞
9	16¼	1¾	14	12	1
10	17½	1⅞	15¼	16	1
12	20½	2	17¾	16	1⅛
14	23	2⅛	20¼	20	1⅛
15	24½	2 3/16	21½	20	1¼
16	25½	2¼	22½	20	1¼
18	28	2⅜	24¾	24	1¼
20	30½	2½	27	24	1⅜
22	33	2⅝	29¼	24	1½
24	36	2¾	32	24	1⅝
26	38¼	2 13/16	34½	28	1⅝
28	40¾	2 15/16	37	28	1⅝
30	43	3	39¼	28	1¾
32	45¼	3⅛	41½	28	1⅞
34	47½	3¼	43½	28	1⅞
36	50	3⅜	46	32	1⅞
38	52¼	3 7/16	48	32	1⅞
40	54½	3 9/16	50¼	36	1⅞
42	57	3 11/16	52¾	36	1⅞
44	59¼	3¾	55	36	2
46	61½	3⅞	57¼	40	2
48	65	4	60¾	40	2

These Drilling Templates are in multiples of four, so that fittings may be made to face in any quarter, and bolt holes straddle the center line.

Bolt holes are drilled ⅛ inch larger than nominal diameter of bolts.

These methods have been superseded naturally by the shrunk and rolled joint, whereby the flange is bored out to a shrink fit, then heated and placed on the pipe, after which the pipe is expanded by a large power roller expander until it not only fits the barrel of the flange but the metal of the pipe flows into the corrugations in the hub of the flange not shown, but usually one corrugation in each half of flange. The pipe is then swung in double-ended lathe and both flange and pipe faced off parallel.

This type of joint was used very successfully on high pressure steam, exhaust and other low pressure service, but the advent of the Vanstone joint has superseded it for high-pressure steam service, as it is a better and safer job, eliminating the chances of leaks between the flange and pipe. Now the shrunk joint is used mostly on exhaust, condenser and water lines.

Vanstone Flanged Joints. This is the only first class commercial joint which is dependable and requires no attention or renewing beyond an occasional gasket (Fig. 14-*B*).

It is made by rolling over the end of the pipe, in front of the flange, until at right angles to axis of pipe. This lap is then faced on front and edge and acts as bearing for gasket in making the joint. This joint can be used in connection with a flange of a valve or fitting as well as two pieces of pipe. The flanges act merely as two swivelling collars to hold the pipe together, the flanges permitting turning for matching fitting, valve or other flanges in any position.

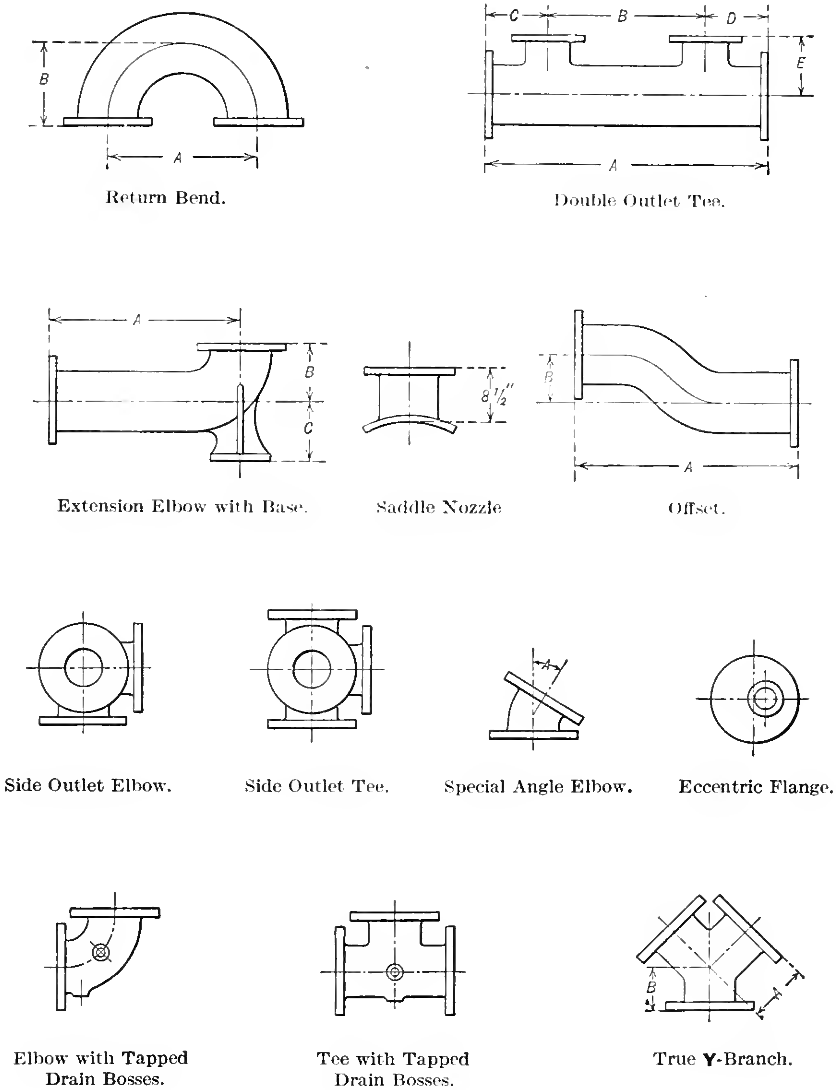


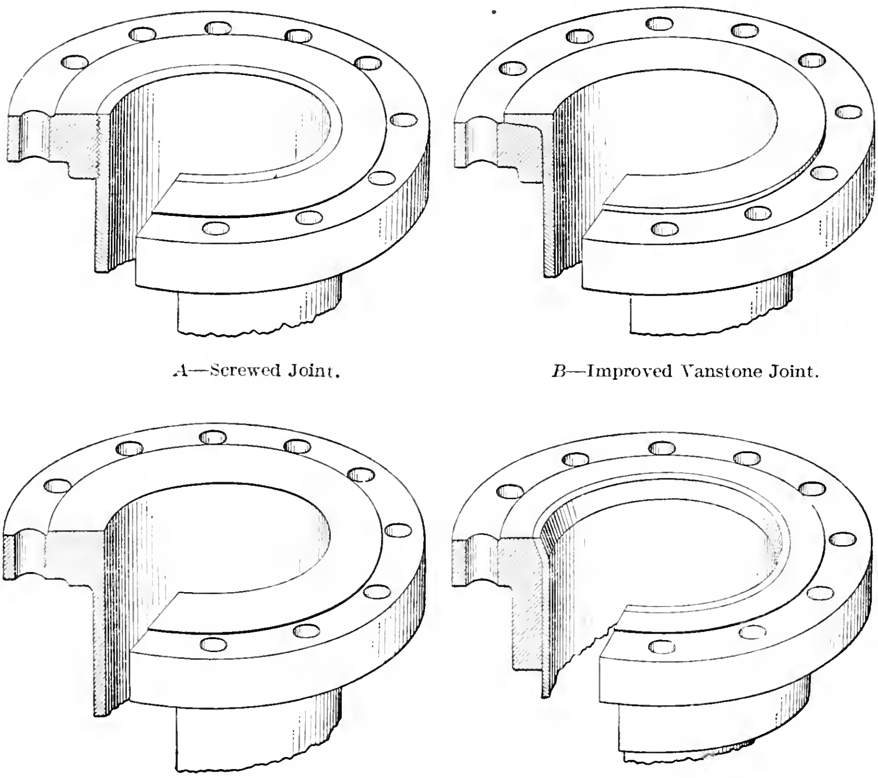
FIG. 13. STANDARD CAST-IRON FLANGED FITTINGS.

Special Patterns. Working Pressure, 150 lb.

NOTE.—See statement in paragraph just preceding Table 16.

The Vanstone joint is adaptable for all classes of service, including steam, gas, air and water, for pressures up to 1,000 pounds. It can be furnished, male and female, when required.

Welded Flanges. In making this joint, which is the most expensive of all forms, a forged steel flange is welded to the pipe end and finished true and square as described above (Fig. 14-*C*).



A—Screwed Joint. *B*—Improved Vanstone Joint.

C—Welded Joint. *D*—Shrunk Joint.

FIG. 14. METHODS OF CONNECTING FLANGES TO WROUGHT PIPE.

Crane Company makes the following cost comparison of these four joints as made by them: (1) screwed = \$19.00; (2) Cranelap (improved Vanstone) = \$23.00; (3) shrunk joint = \$24.00 and (4) Craneweld = \$34.00.

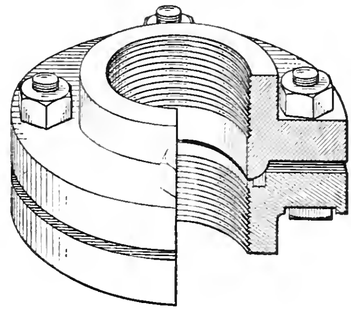
The following methods of *finishing the flange faces* in order to make a tight joint when the bolts are drawn up are used and described by the above company as follows:

Methods of Facing Flanges. (*a*) Plain straight face; (*b*) raised face smooth finish for gaskets; (*c*) raised face finished for ground joint; (*d*) tongue and groove; (*e*) male and female; (*f*) plain face corrugated; (*g*) plain face scored; (*h*) ball shape for ground joint. See Fig. 15.

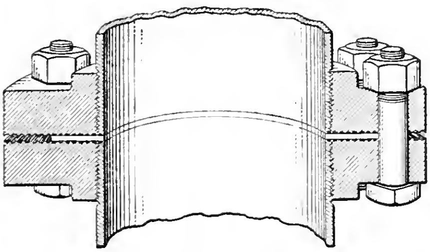
"*Plain Straight Face.* This type of joint has the entire face of the flange faced straight across and uses either a full face or ring gasket. It is commonly employed for pressures less than 125 pounds on steam and water lines. The best results are obtained by using a fairly thick gasket, so that the gasket will have sufficient pressure exerted on it by the bolts to make a tight joint, before the outside edges of the flanges meet. The full-faced gasket is preferred by some, because it may be installed

more readily, and is more likely to be concentric with the bore of the flange than that of a ring gasket, but it has no further advantages.

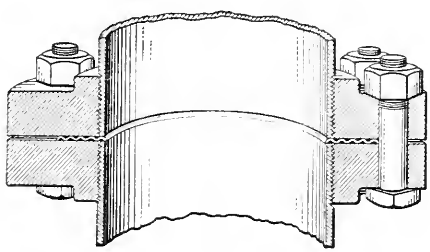
A ring gasket, properly proportioned and correctly installed, will make just as tight a joint as a full-faced gasket, at considerably less expense and with less pulling up on the bolts.



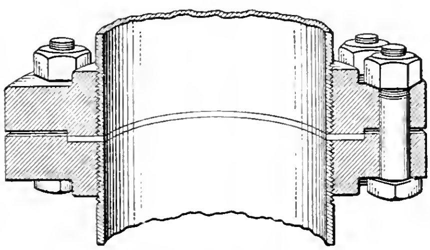
Two-Part Flange Union.



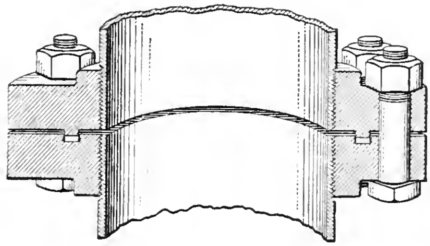
Corrugated Faces with Rubber Gasket.



Smooth Faces with Corrugated Copper Gasket.



Male and Female Faces with Rubber Gasket.



Tongued and Grooved Faces with Rubber Gasket.

FIG. 15. TYPICAL FLANGE CONNECTIONS.
Facing and Gaskets.

"*Raised Face Smooth Finish for Gaskets.* This type of face is made raising the face of the flange between the bore and inside of the bolt holes, $1/32''$ to $1/16''$ above that of the remainder of the flange.

"This type of joint is most satisfactory on high pressure steam lines and it is the most general on the market to-day.

"With this style of face ring gaskets are employed, and a greater pressure per square inch of gasket is obtained by pulling up on the bolts than would be obtained with similar bolts on a full-face gasket.

"The raised face prevents the touching of the outside edges of the flanges, and the entire pressure exerted by the bolts is transmitted to the gasket, which gives a maximum efficiency and resistance against leakage.

"*Raised Face Ground Joints.* This style of face is identical with that employed when gaskets are used, excepting that the raised face is ground to an absolute metallic joint. This eliminates the use of gaskets.

"This style of joint was popular before a satisfactory gasket material was found, and was employed considerably on superheated steam lines. There having been placed on the market gaskets which are employed for temperatures as high as 800° F., the successful use of these gaskets has to a considerable degree reduced the number of ground joints used in steam lines.

"*Tongue and Groove Flanges.* This style of joint is very popular. On high pressure water, gas or air lines the gasket area is reduced to a minimum, thereby increasing the pressure on the gasket further than on any other style of gasket construction. The area of the gasket in this joint is considerably less than would be necessary if the gasket were not protected from blowing outward or squeezing inward, as is the case with unprotected gaskets. The installation of a joint of this kind is more expensive than when raised face is used. The disassembling, for replacing of a valve or fitting, is also expensive unless lines are made so that they may be sprung readily.

"For hydraulic and ammonia lines there is no better joint.

"*Male and Female.* Male and female joints are employed on steam lines having high pressures, and were also employed on hydraulic lines where a cup-shaped gasket was used. They are still being employed for services of this kind, for the reason that the gasket is retained from blowing out, but the width of the gasket is greater than when a tongue and groove is used. This style of joint is as expensive to install and disassemble as a tongue and groove joint.

"*Plain Face Corrugated Joints.* This style of joint is nothing more than a plain face straight flange upon which concentric curves have been cut with a round nosed tool. On some types of installations a face of this kind is necessary, as the corrugations have a tendency to prevent the gaskets from blowing out, particularly when the flow in the pipe line is of a nature which requires the use of exceptionally thick gaskets.

"*Plain Face Scored.* This type of joint is made by using a plain straight flange with scores upon the face consisting of concentric rings made with a diamond pointed tool. On oil or acid lines, where the gaskets must be of lead, a joint of this kind gives the best satisfaction, as the lead gasket squeezes into the scores and assists in maintaining a tight joint without any undue strain on the bolts and flanges.

"*Ball Shaped Flanges for Ground Joints.* The use of flanges having inserted parts and non-corrosive rings is increasing every year. This is due to the fact that screwed unions of this type are being made to this construction.

"The success of these types of unions has induced manufacturers to make flange unions of similar construction. The elimination of the gasket is construed by many engineers as an improvement. The ball joint allows a reasonable misalignment of the piping, thereby reducing the breakage of flanges due to that cause to a minimum. In high class installations where the faces of the flanges are machined with the axis of the pipe, a ball joint union would be of no advantage, but in cases where there is considerable settlement of the building or other misalignment in the piping this style of face meets with considerable success."

TABLE 20

REINFORCED VANSTONE JOINTS

(See Fig. 16

Pipe Size, In.	*D* In.	*T* In.	*T'* In.	Pipe Size, In.	*D* In.	*T* In.	*T'* In.	Pipe Size, In.	*D* In.	*T* In.	*T'* In.
4	4 1/2	1/4	5/16	9	9 5/8	3/8	15/32	15	15	3/8	1/2
5	5 9/16	1/4	3/8	10	10 3/4	3/8	1/2	16	16	3/8	1/2
6	6 5/8	9/32	13/32	12	12 3/4	3/8	1/2	18	18	3/8	1/2
7	7 5/8	5/16	7/16	14	14	3/8	1/2	20	20	3/8	1/2
8	8 5/8	11/32	7/16								

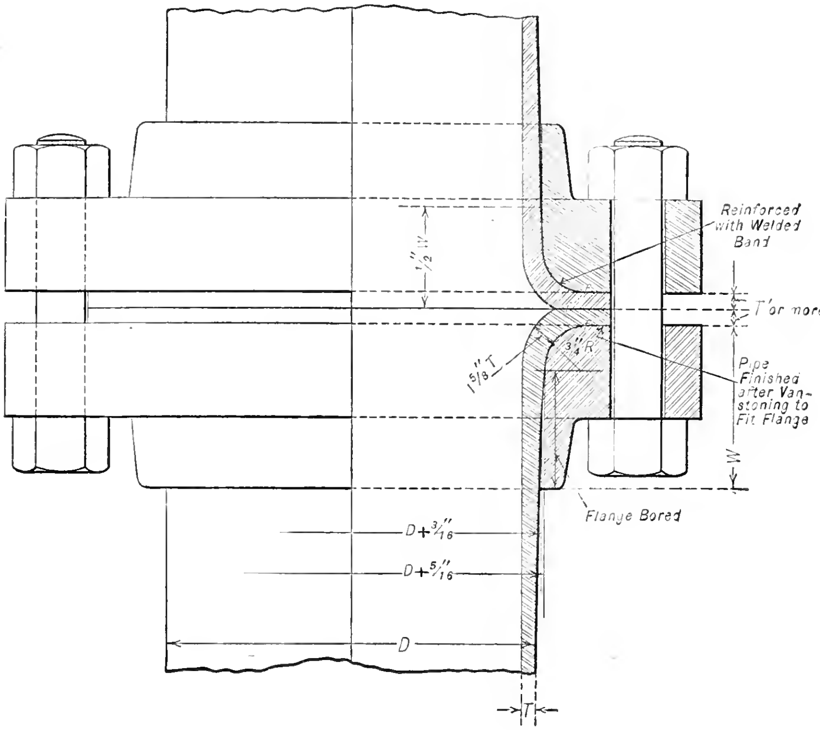


Fig. 16. Reinforced Vanstone Flanged Joint. (*M. W. Kellogg Co.*)

(See Table 20.)

VALVES

Commercial valves, like fittings, are usually either made with *screwed* or *threaded ends*, or else are *flanged* for bolted connection to corresponding flanges on the pipe or fitting, in which case the flange and bolting arrangement should conform with the *American Standard* for flanged fittings.

Materials Used in Valves. The material used for valves of small size is generally *brass*, but in the larger sizes either *cast iron*, *cast steel* or some of the *steel alloys* are employed. Practically all iron or steel valves intended for steam or water work are brass-mounted or trimmed, while valves for acids, ammonia and corrosive gases are of iron throughout.

Pressure Requirement. Valves for *general service* are generally designed for *standard* or *extra heavy* service, the former being used up to 125 lb. and the latter up to 250 lb. steam working pressure, although most manufacturers also make a valve for *low pressure* up to 25 lb. steam and for *medium pressure* up to 175 lb. steam working pressure. In practically all cases these valves are tested at a cold water pressure of twice the steam pressure, and may be used on water lines with a pressure 40 per cent in excess of the allowable rated steam pressure. Some manufacturers rate their standard valves at 150 lb. steam working pressure instead of 125 lb. as stated above.

In addition to valves for general service there are valves for special service requirements, such as heating systems which are used only on low pressures and are often made of special shape as shown in Figs. 29 to 35.

Types of Valves. The types of valves commercially available are almost unlimited, as practically every requirement for controlling the flow of water, steam or gas has been met in the design of some sort of valve especially adapted to the service in question. Only the more common types are considered here, such as *gate valves* or straightway valves, *globe valves, angle valves, check valves,* and a few *automatic valves*, such as *reducing* and *back-pressure* valves

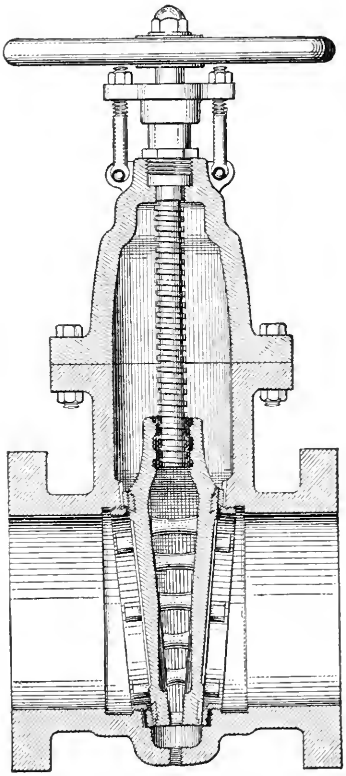


FIG. 17. IRON BODY GATE VALVE WITH RENEWABLE SEAT RINGS.

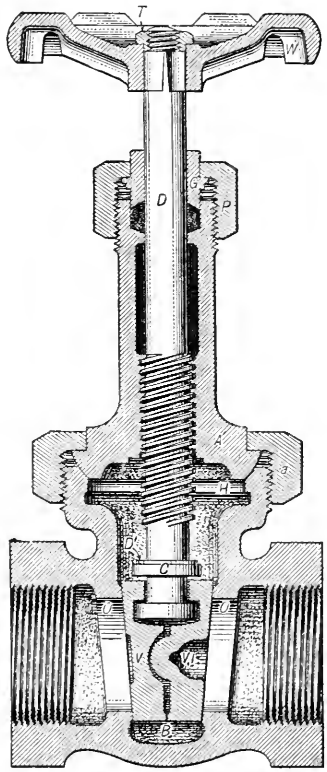


FIG. 18. ALL-BRASS GATE VALVE WITH UNION BONNET.

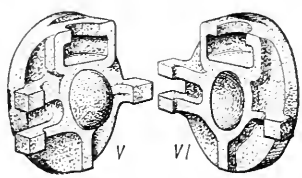


FIG. 19. SPLIT WEDGE.

Gate Valves. The gate valve (Figs. 17 and 18) is probably the most commonly used of all valves, and is to be desired in all cases except those in which a throttling action is necessary, when a globe or angle valve may be required. These valves are made in sizes from 2″ to 12″ with screwed ends, and from 2″ to 30″ with flanged ends, using *cast-iron* body, bonnet and wedge, or discs, but are *brass- or bronze-mounted* with removable seats, disc rings, and stems.

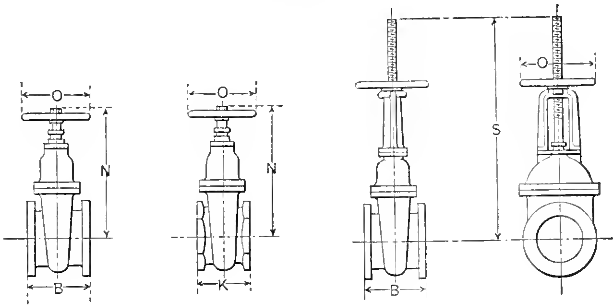
TABLE 21

DIMENSIONS OF IRON BODY GATE VALVES

Inside Screw, Outside Screw and Yoke, Iron Body, Brass-Mounted

125 and 25 Pounds Steam Working Pressures

(Best Mfg. Co.)



125 Pounds Pressure											25 Pounds Pressure				
Size	*B*	*K*	*N*	*S*	*O*	Size	*B*	*N*	*S*	*O*	Size	*B*	*N*	*S*	*O*
2	7	5 7/16	11¾	14	6½	16	16	42¾	74¾	22	12	11	32⅝	55	16
2½	7½	5⅞	12¾	15¾	6½	18	17	48¾	86	24	14	13½	36¼	62½	16
3	8	6⅛	14¼	18½	7½	20	18	52½	91	24	16	14	41	71⅝	18
3½	8½	6½	15¼	20¾	7½	22	19	55½	100	27	18	14½	44¾	79¾	18
4	9	6⅞	16¼	23½	9	24	20	62	109	30	20	15½	48¾	87½	20
4½	9½	7¼	17⅝	24¾	9	26	23	65⅞	117½	30	22	16½	52⅛	94⅞	20
5	10	7 5/16	19	28	10	28	26	70	125	36	24	17	56½	103⅞	22
6	10½	7¾	20¾	31¾	12	30	30	75½	133	36	26	18½	61⅛	112	22
7	11	8¼	23	37¼	12	36	36	83	158½	42	28	20	65⅜	122¼	24
8	11½	8 11/16	26	41	14						30	21	68⅞	128½	24
9	12	9¼	28	44¼	14						36	24	81½	151½	30
10	13	9⅛	30¼	49½	16						42	27	94	178	36
12	14	11⅝	35¼	57¼	18						48	30	111¾	204	48
14	15	......	39¼	66½	20						54	33	127½	236	*
15	15	......	41⅛	69¾	20						60	36	138¼	258	*
												66, 72	Dimensions on Application		

* Geared.

For diameter, drilling, and thickness of flanges, see standard flanges.

These valves may have either a *rising or non-rising spindle* and are usually provided with a yoke for guiding the spindle when same is of the rising type. If of the non-rising type (Fig. 17), then the spindle must turn and screw into the wedge nut as the wheel is revolved, in which case it is not apparent to the eye whether the valve is shut or open. The wedge may be of the *solid*, or the double or *split* type, and should always be designed with a slight taper so that it will close tight against the tapered seats, which are usually furnished with renewable seat rings. The solid wedge or split discs should not drag over the seats in opening or closing, but move straight up or down to avoid scoring. The top of the wedge should seat against the bonnet when the valve is wide open so that it may be packed at the spindle stuffing-box while open and under pressure.

Dimensions of iron body, brass-mounted gate valves are given in Tables 21 and 22.

All-brass gate valves are made in sizes from ¼″ to 3″ in diameter, and generally have screwed ends as shown in Fig. 18, in which the bonnet A is secured to the body of the valve by the union nut a. The spindle D passes through the stuffing-box, which is made tight by the nut P and

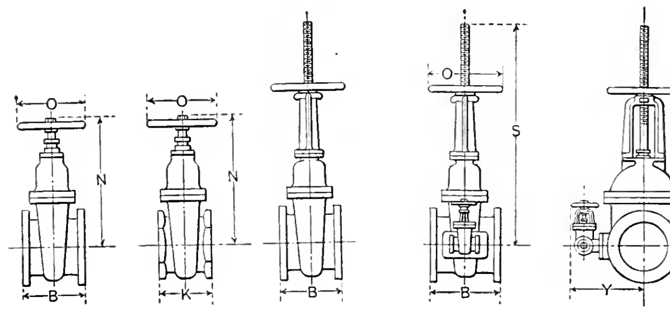
TABLE 22

DIMENSIONS OF IRON BODY GATE VALVES

Inside Screw, Outside Screw and Yoke, Iron Body, Brass-Mounted

175 and 250 Pounds Steam Working Pressures

(Best Mfg. Co.)



175 Pounds Pressure							250 Pounds Pressure						
Size	*B*	*K*	*N*	*S*	*O*	*Y*	Size	*B*	*K*	*N*	*S*	*O*	*Y*
2.........	7½	5½	11⅜	14	6½	....	2.........	8½	7	10½	13¾	6½	....
2½........	8	6	12⅜	15½	6½	....	2½........	9½	8	12⅞	16	7½	....
3.........	9½	7¼	14 3/16	18⅜	7½	....	3.........	11⅛	9	14⅝	19½	10	....
3½........	10	7½	14¾	20½	7½	....	3½........	11⅞	10	15½	22	10	....
4.........	10½	7¾	16¼	23¾	9	....	4.........	12	11	17¾	24½	12	....
4½........	11	8¼	17	25	9	....	4½........	13¼	12¼	18¾	27	12	....
5.........	11½	8½	19	28⅛	10	....	5.........	15	13½	20¼	29¾	14	12⅝
6.........	12	8¾	21⅛	31⅞	12	14 5/16	6.........	15⅞	15⅞	23	34⅛	16	13
7.........	12½	9¼	22⅞	35¾	12	15⅛	7.........	16¼	16¼	24¾	38	16	14⅛
8.........	13½	10	25⅞	40¾	14	15⅞	8.........	16½	16½	28¾	42¾	20	15⅞
9.........	14	10¾	27½	44¼	14	16 9/16	9.........	17	17	30½	47	20	16⅜
10.........	15	11½	30⅛	49¾	16	17⅜	10.........	18	18	33¾	52¾	22	16⅞
12.........	16	12½	33¾	56⅞	18	18⅝	12.........	19¾	....	37¼	60	22	19⅞
14.........	18	....	38½	64½	20	19¾	14.........	22½	....	42¾	67¾	24	20⅝
15.........	18¾	....	41	68¾	20	20½	15.........	22½	....	42¾	67¾	24	20⅝
16.........	19½	....	44¾	74½	22	22¾	16.........	24	....	....	75¼	27	25¼
18.........	21	....	47⅛	82¼	24	24¼	18.........	26	....	....	82¼	30	26½
20.........	22½	....	51⅞	90¾	24	27 9/16	20.........	28	....	....	91½	30	30½
22.........	24	....	55¾	98⅝	27	28 9/16	22.........	29½	....	....	101	36	32¼
24.........	25½	....	59⅞	107	30	30½	24.........	31	....	....	109	36	33
26.........	29½	....	....	118	36	33	26.........	36	....	....	118	42	35
28.........	33½	....	....	125	36	36	28.........	41	....	....	125	42	38
30.........	37½	....	....	133	36	39	30.........	45	....	....	133	42	41

For diameter, drilling, and thickness of flanges, see standard flanges.

follower G which bears against the packing. The wheel W is secured by nut T to the top of the spindle, and the shoulder C and hub at the lower end turn in the split wedge V-V_1, which seats against the tapered faces at O. The finished face D' back seats at H, when valve is wide open so that it may be packed under pressure.

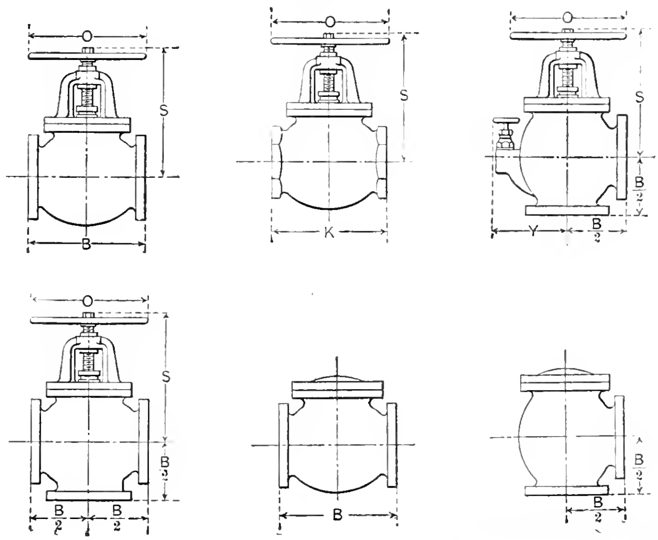
Globe, Angle, and Cross Valves. Globe, angle, and cross valves, like gate valves, are made with *iron bodies* and are usually *brass-mounted*. The principal dimensions are given in Table 23, and it will be noted (Fig. 20) that in all cases a yoke is used for guiding the spindle, which is of the *rising* type, and designed to send the valve disc to its seat against the full line pressure, which should always be on the under side of the disc when the valve is closed. The wheel of this valve is usually fastened to the spindle with a nut as shown, and moves up and down with it, making it necessary to allow ample clearance for the entire wheel. It will be noted that the seat is re-

TABLE 23

DIMENSIONS OF GLOBE, ANGLE, CROSS, AND CHECK VALVES, IRON BODY, BRASS-MOUNTED

125 and 250 Pounds Steam Working Pressures

(*Best Mfg. Co.*)



125 Pounds Pressure						250 Pounds Pressure							
Size	*B*	*B*/2	*K*	*S* Open	*O*	Size	*B*	*B*/2	*K*	*S* Open	*O*	*Y*	Size of By-pass
2	8	4	5¾	10¾	6½	2	10½	5¼	9½	13¾	7½	....	...
2½	8½	4¼	6⅝	11¼	6½	2½	11½	5¾	10¾	14½	10	....	...
3	9½	4¾	7½	12¾	7½	3	12½	6¼	11¾	17½	10	....	...
3½	10½	5¼	8¼	13	7½	3½	13¼	6⅝	12¼	17½	10	....	...
4	11	5½	9⅜	15¼	9	4	14	7	13	19½	14	....	...
4½	12	6	10¼	15¼	9	4½	15	7½	14	19½	14	....	...
5	13	6½	11¼	17¼	10	5	15¾	7⅞	15	21½	16	....	...
6	14	7	12½	19	12	6	17½	8¾	16½	25	18	....	...
7	16	8	14	20½	14	7	19¼	9⅝	18¼	26¼	20	....	...
8	17	8½	16	23¾	16	8	21	10½	20	29½	24	12⅝	1½
10	20	10	18¾	28	18	10	24½	12¼	23¼	33½	27	14¼	1½
12	24	12	22	34	20	12	28	14	....	39	30	16⅝	2
14	28	14	....	38½	24	14	33	16½	....	42	36	18	2
15	30	15	....	38½	24	15	33	16½	....	42	36	18	2
16	32	16	....	41½	27								

For diameter, drilling, and thickness of flanges, see standard flanges.

movable, and the valve disc has an extended spindle moving in a suitable guide, formed by three ribs.

All-brass globe, angle, and *cross valves* are made in sizes from ⅛″ to 4″ diameter and generally have screwed ends as shown in Fig. 21. Like the all-brass gate valve, the valve shown is a *Powell White Star* valve with union bonnet, and has a removable disc holder *H* held in place by pin at *P*. The composition disc *V* is secured by nut *S* and may be readily replaced whenever it becomes worn.

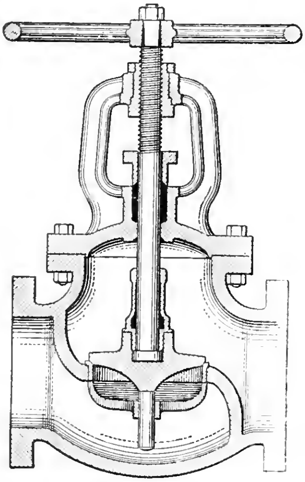


FIG. 20. IRON BODY GLOBE VALVE—BRASS-MOUNTED.

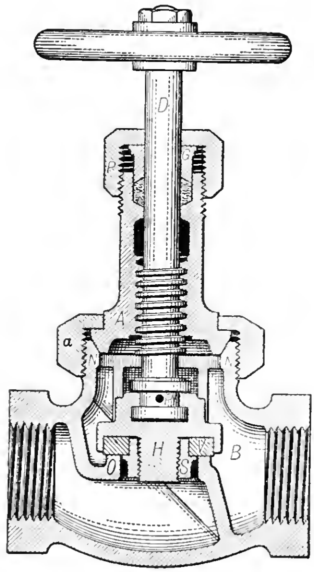


FIG. 21. SECTION OF ALL-BRASS GLOBE VALVE WITH RENEWABLE DISC.

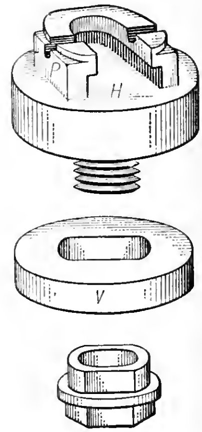


FIG. 21a

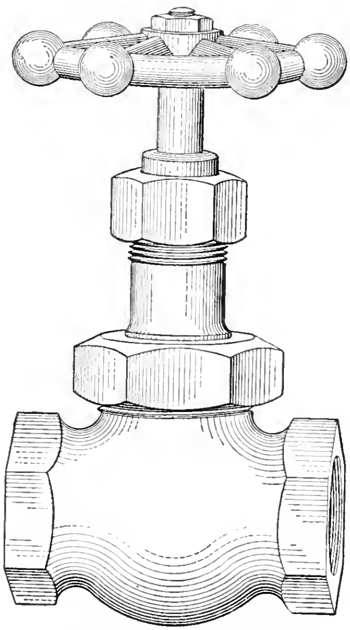


FIG. 21b. EXTERIOR VIEW. (*Powell White Star.*)

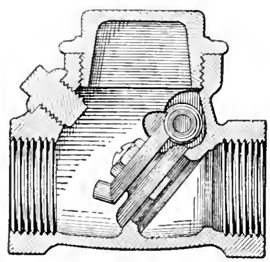


FIG. 22. SWING CHECK VALVE.

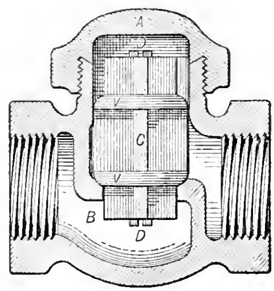


FIG. 23. LIFT CHECK VALVE.

Automatic Valves. Automatic valves include many varieties, and, in fact, practically any valve may be made automatic by the attachment of the necessary weighted levers, springs, or pressure diaphragms.

Check Valves. The simplest of all automatic valves is, of course, the check valve, which opens whenever the unbalanced pressure below the valve is sufficient to lift it and closes when this pressure fails or flow starts in the opposite direction. These valves may be either hinged to swing as in Fig. 22, where provision is made for regrinding by removing the small plug directly in line with the axis of the valve; or a dead weight disc, moving vertically, may be used as in

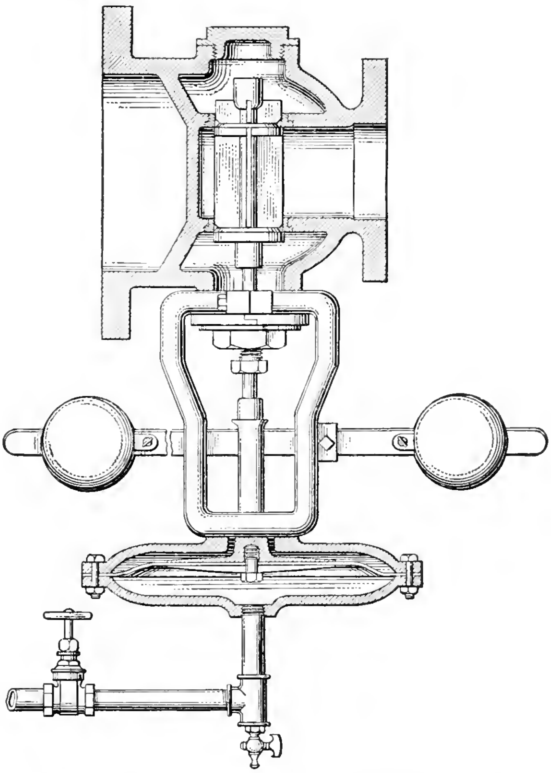


Fig. 24. Pressure Reducing Valve Monash Class C-1.

Fig. 23, where a reversible disc is shown. The valve in Fig. 22 can also be used in a vertical line, while the one in Fig. 23 cannot.

Pressure Reducing Valves. The reducing valves, which are very largely used in heating work whenever steam is generated at high pressure and then used at low pressure in the radiators, are either of the diaphragm and weighted lever type (Fig. 24), or else use a compression spring in place of the weight and lever. A large diaphragm is necessary whenever a reduced pressure of 10 lb. or less is to be maintained.

When the reduced pressure is above 10 lb. it is usually possible to use a piston moving in a cylinder cast integrally with and opening into the low pressure side of the valve. The reduced pressure, operating on this piston, acts to close the valve in opposition to a weighted lever or spring which tends to open the valve.

The *Monash* reducing valve shown in Fig. 24 has the inlet smaller than the outlet, and

uses a double or balanced valve which overcomes the tendency of the high pressure steam to either blow a single valve open or force it closed. This valve moves up or down under the influence of the low or reduced pressure steam, which acts upon under side of the diaphragm attached to the bottom of the valve spindle. Connection to this chamber is made by the small pipe shown at the bottom of the figure. This connection must be made to the low pressure main at least 15 ft. from the valve if steady operation is to be secured. By proper adjustment of the two weights on the lever bar, any desired low pressure up to 10 lb. is readily secured and automatically maintained. If no steam is allowed to enter the line leading to the under side of the diaphragm the

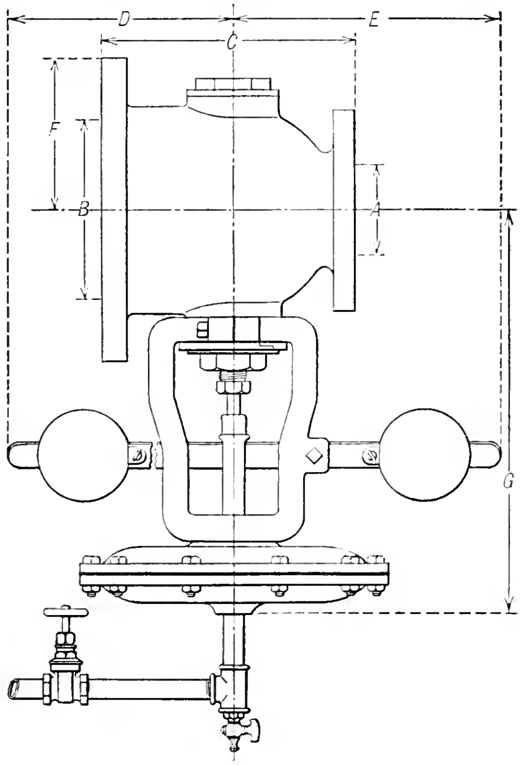


TABLE 24

MONASH CLASS C-1 PRESSURE REGULATING VALVES

ALL DIMENSIONS GIVEN IN INCHES

	1¼x2½	1½ x 3	2 x 4	2½ x 5	3 x 6	3½ x 7	4 x 6	4 x 8	5 x 10	6 x 12	8 x 14	8 x 16
A	1¼	1½	2	2½	3	3½	4	4	5	6	8	8
B	2½	3	4	5	6	7	6	8	10	12	14	16
C	6⅜	7⅜	8 5/16	9	10¼	10¾	10 7/16	11⅜	13½	15	17 9/16	17⅝
D	12½	12½	13 13/16	15⅜	17	18 9/16	17	20⅛	21 11/16	23⅛	27	27
E	15½	15½	16 3/16	17⅝	19	20 7/16	19	21⅞	23 5/16	24⅞	33	33
F	3½	3¾	4½	5	5½	6¼	5½	6¾	8	9½	10½	11¾
G	13⅞	14⅝	15⅛	16½	16 13/16	17 13/16	16 13/16	18 7/16	20⅜	21 5/16	23⅝	23⅝

NOTE.—Sizes up to and including 2 x 4 inch are made with screwed inlet and flanged outlet. All larger valves have both ends flanged. For 125 lb. steam working pressure and not over 10 lb. reduced pressure.

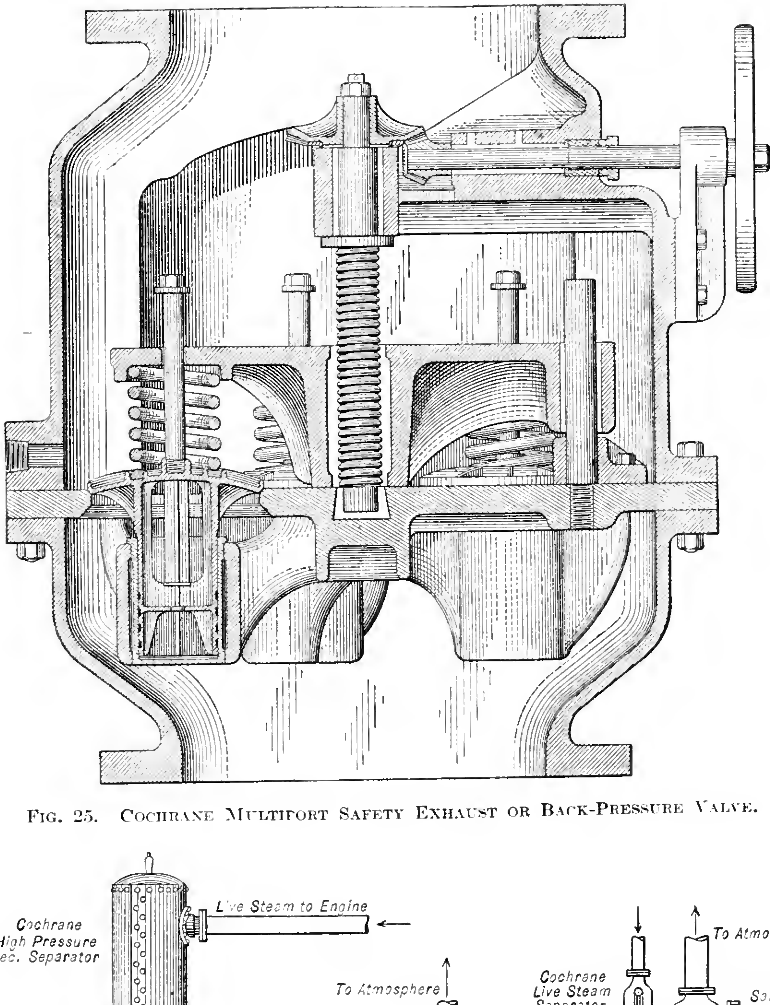


FIG. 25. COCHRANE MULTIPORT SAFETY EXHAUST OR BACK-PRESSURE VALVE.

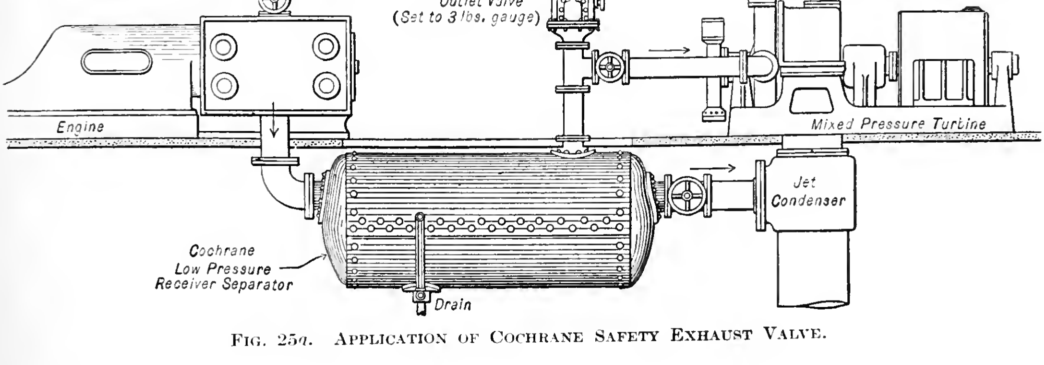


FIG. 25a. APPLICATION OF COCHRANE SAFETY EXHAUST VALVE.

valve will remain wide open under the influence of the weighted lever, and will pass high pressure steam like any other valve. The valve shown is designed for 125 lb. working pressure.

Back-Pressure and Non-Return Valves. There are a great variety of automatic valves which act as *back-pressure* or *non-return* valves (1) to prevent the return of steam or water through them or (2) to maintain a certain predetermined pressure before they will open and relieve a dangerous or undesirable pressure which might develop in the system.

The *Cochrane Multiport* safety exhaust or back-pressure valve (Figs. 25 and 25*a*) is the most improved type of relief valve now on the market. It is so designed as to permit of the closest regulation of the back pressure which is to be maintained. All springs are under the control of a single pressure plate, which is operated by gears as shown by means of an exterior hand wheel. Each valve is equipped with dash-pot on guides so that it will not slam in closing and will travel straight and true in order to give an even bearing over its seat.

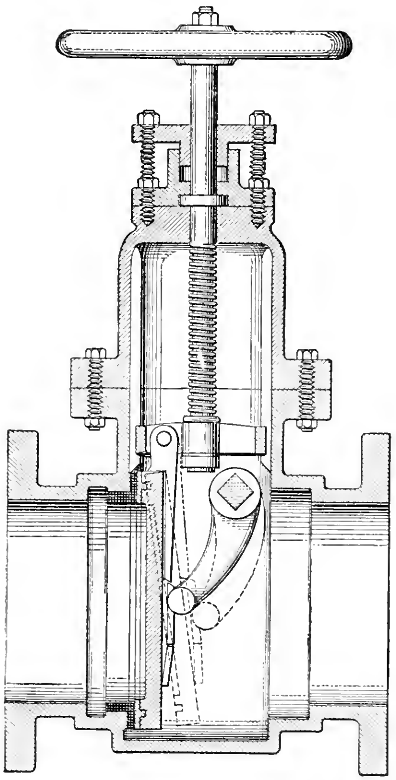


Fig. 26. Interior of the Erwood Swing-Gate Valve. The Dotted Lines Show the Swing-Gate Partly Open.

Such valves may act merely as weighted check valves or back-pressure valves, or they may be of the gate or globe type, and combine the automatic or self-closing feature with the positive closing feature of ordinary stop valves.

A valve of this sort, known as the *Erwood Swing-Gate*, intended primarily for service on exhaust-steam connections, although applicable for non-return service between a boiler and steam header, is shown in Fig. 26, and its application illustrated in Fig. 26*a*. The gate of this valve is held to its seat by an adjustable external spring, operating on a pivoted lever arm, the axis of which passes through the body of the valve, and holds the valve against its seat under whatever pressure may be desired. The steam pressure operates on the other side of the gate, causing it to swing open, unless the pressure is insufficient, or unless a reversal in flow occurs, as in the case of the failure of one of a battery of boilers, when flow from the header into the ruptured boiler would be cut off. This valve may be installed in any position and operated at any angle, but, of course, must be used only as a one-way valve.

In service (Fig. 26*a*) the valve is used as at *A* to prevent an open heater or receiver from flooding the engine, and again, as at *B*, on the atmospheric exhaust line from the heater, to maintain a slight back pressure on the system and supply steam for heating through the line *C*.

Blow-off Valves and Cocks. Special valves or cocks must be used on certain lines such as blow-off lines from power boilers, where the nature of the service rather than the pressure determines the type of valve to be used. A *blow-off valve* (Fig. 27) made by the *Lunkenheimer Co.*, and typical of this class, illustrates the development of this sort of equipment providing for all contingencies and probable replacements, as a consideration of its features will indicate. The plug fits snugly in a separate and easily removable bronze casing, which can be readily replaced when worn. Any accumulation of scale or sediment that might remain on the seat before the disc is brought in contact with same is washed off by the water which passes around the plug when seating.

Ordinary *plug cocks* (Fig. 28) on blow-off lines are satisfactory for low pressure heating work, and specially designed cocks or *asbestos-packed cocks* are often used on high pressure blow-off lines. These cocks open and close with a quarter turn.

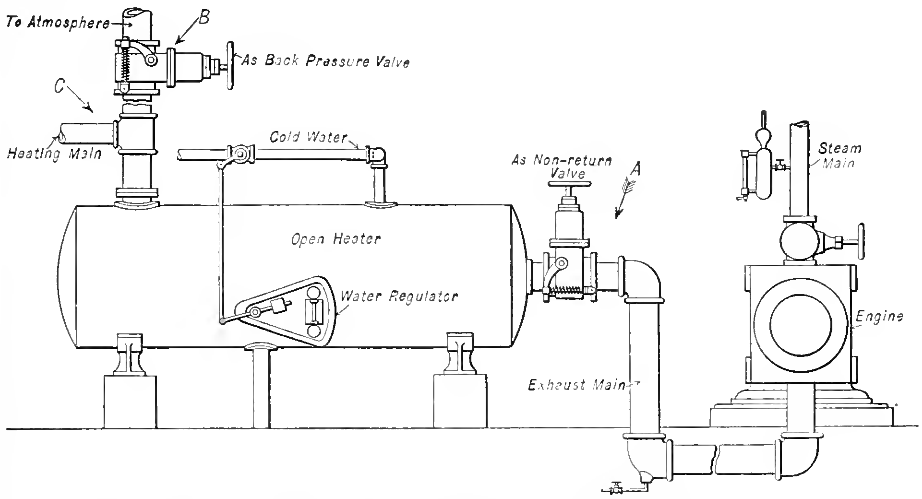


FIG. 26*a*. APPLICATION OF ERWOOD SWING-GATE VALVE.

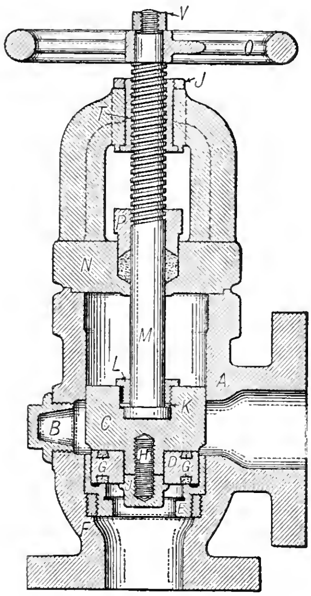


FIG. 27. BLOW-OFF VALVE.
(*Lunkenheimer.*)

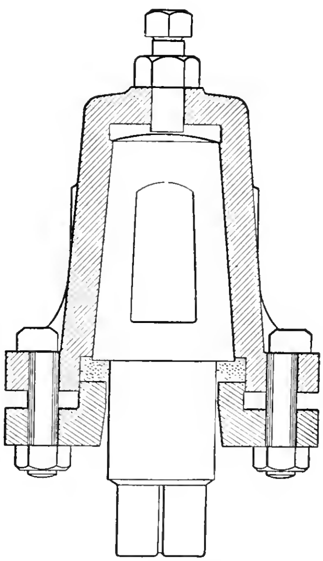


FIG. 28. PLUG COCK.
(Asbestos Packed.)

Identification of Valve. A *directory of valves* or a proper *means of identification* of the lines controlled by them should be required in all systems wherein any possibility of confusion as to which valve to operate to affect a certain line may arise.

Valves for Heating Service. A special class of valves is required for controlling steam and water radiators and they are briefly considered here. These valves are of brass, usually of the angle type, modified to suit the service requirements, and are often arranged with graduated heads and lever handles in order to indicate the relative opening of the valve port in any position.

Steam Radiator Valves. The most common type of steam radiator valve is the *angle pattern* (Fig. 29), equipped with wood wheel, ball-joint union, and removable composition disc. These valves range in size from ½″ to 2″, and may be furnished with or without the union connection, which in general should always be specified, as it facilitates disconnection. In addition to the *direct angle pattern,* this valve may be obtained in the *corner* or *offset corner* pattern (Fig. 30), as well as in the *straight offset* or *offset globe* pattern. Dimensions of the angle and offset corner valves are given in Table 25, and the advantages of the corner and offset patterns will be at once apparent in simplifying radiator connections.

Hot-Water Radiator Valves. The above valves* may be used for hot-water heating service by drilling a $^1/_{16}$″ diameter hole through the web forming the seat so as to permit sufficient circulation to take place to prevent freezing when valve is closed. A much simpler form of hot-water radiator valve, of the *butterfly type* (Fig. 31), is manufactured and serves the purpose of controlling the flow, although it does not close perfectly water-tight. The foot lever may be omitted and a wooden wheel substituted for same.

The *quick-opening* (Q.-O.) *hot-water radiator valve* (Fig. 32), in which a quarter turn of the handle revolves a cylindrical or conical shell so that the port through same registers with an opening in the valve body, is more generally used on water work than the

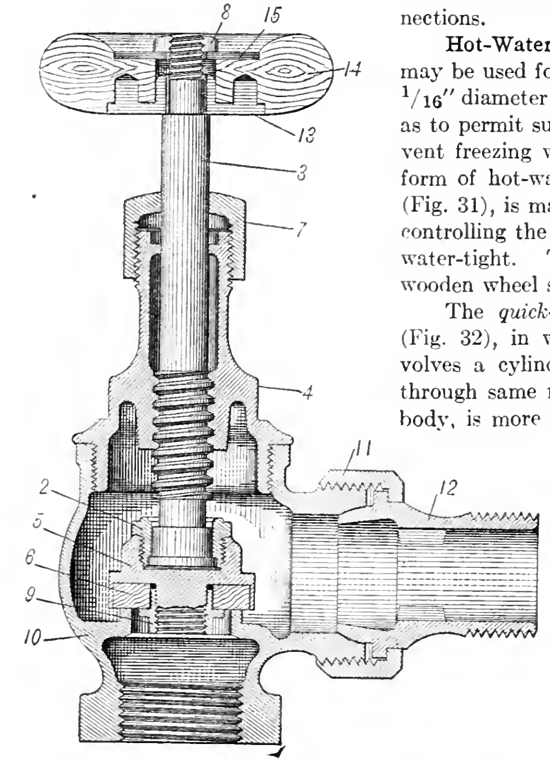


FIG. 29. STEAM RADIATOR VALVE—ANGLE PATTERN (*Jenkins Bros.*).

Description of Parts.

1. Wheel	6. Disc	11. Union Nut
2. Lock Nut	7. Waste Nut	12. Union Nipple
3. Spindle	8. Wheel Nut	13. Bottom Plate
4. Bonnet	9. Disc Nut	14. Wood Wheel
5. Disc Holder	10. Body	15. Top Plate

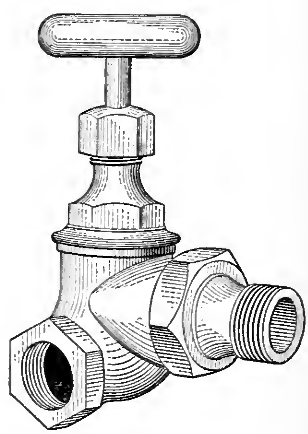


FIG. 30. OFFSET CORNER VALVE WITH MALE UNION, RIGHT HAND.

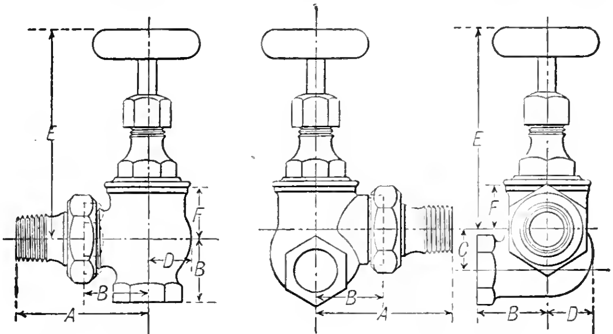
butterfly type of valve shown above. The valve shown has a conical shell and globe-shaped body, which helps materially to do away with the sticking of the shell, since only a small

* NOTE.—When these valves are used on forced hot-water work it is not customary to drill a circulation hole in same. For hot-water service a follower is always supplied for securing the packing in the stuffing-box.

portion of the shell comes in contact with the body at the top and bottom, and at a narrow vertical strip on either side where a gate is formed for closing the water-way. The tapering shell permits of taking up of any wear which may occur in the valve. The spring in the bonnet or neck of the valve holds the conical shell up to its seat, and at the same time exerts a downward pressure on the small rubber washer which is slipped over the stem and held within the chamber in the cap of the valve. The pressure of the spring expands the rubber gasket so as to provide a self-packing feature.

TABLE 25

DIMENSIONS OF JENKINS BROS. ANGLE RADIATOR AND OFFSET CORNER VALVES



Angle Radiator Valve Offset Corner Valves

Size. Angle Type	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3
A—Center to end of union	2 13/16	3 5/16	3 3/4	4	4 1/2	4 3/4	5 7/8	6 7/8
B—Center to face, screwed end	1 3/8	1 1/2	1 13/16	2 1/16	2 1/4	2 13/16	3 1/4	4 1/4
D—Radius of body	7/8	1 1/16	1 1/4	1 1/2	1 3/4	2 1/4	2 11/16	3 1/4
E—Center of outlet to top of hand wheel	4 3/4	5 1/2	5 1/2	6 1/2	7	8	9	9 1/2
F—Center to top of body	1	1 3/16	1 5/16	1 1/2	1 11/16	2 1/16	2 5/16	2 7/8

Size. Offset Corner Type	1/2	3/4	1	1 1/4	1 1/2	2
A—Center to end of union	3	3 7/16	3 3/4	4 1/8	4 11/16	5 1/8
B—Center to face, screwed end	1 1/2	1 5/8	2	2 1/8	2 9/16	3 1/8
C—Center of outlet to center of inlet	3/4	1	1 5/32	1 3/16	1 1/2	1 11/16
D—Radius of body	7/8	1 1/16	1 1/4	1 1/2	1 3/4	2 3/16
E—Center of outlet to top of hand wheel	4 1/2	5	5 3/8	6 1/4	7	7 1/2
F—Center of outlet to top of body	7/8	1 1/16	1 3/16	1 7/16	1 1/2	2 1/8

NOTE.—Dimensions of offset globe same as offset corner valves. Regular corner valves have no offset. Other dimensions are same as given in table for offset corner valves.

This valve is made in sizes from 1/2″ to 2″ either with or without union as shown in Fig. 32, and should always be connected with bottom inlet and side outlet.

Packless Radiator Valves. The radiator valves so far shown must all be *packed* periodically with cotton or asbestos fiber to prevent steam or water from *leaking out* through the stuffing-box, or to prevent air from *leaking in* through the same place.

A *packless type* of radiator valve has been developed which practically overcomes this difficulty or objection, and in the best designs makes a perfectly tight joint at the spindle against steam water and air. The *Sylphon* packless radiator valve (Fig. 33) makes use of an expanding brass bellows which entirely encases the spindle of the valve so that there is no opening or packed bearing at any point. The bonnet is secured by a union nut as shown, and a renewable composition disc is used as in the ordinary angle valve, of which this is a modification. This valve is an absolutely packless valve.

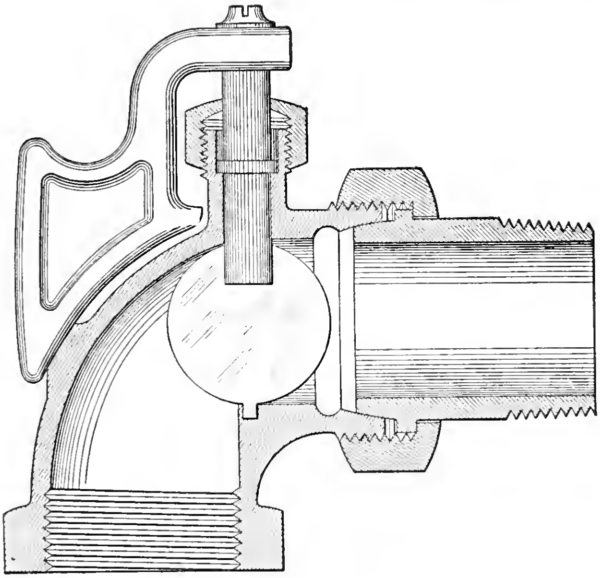


FIG. 31. HOT-WATER RADIATOR VALVE

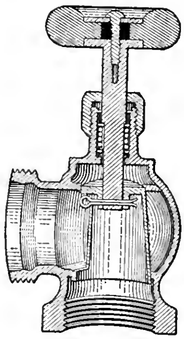


FIG. 32. QUICK-OPENING HOT-WATER RADIATOR VALVE.

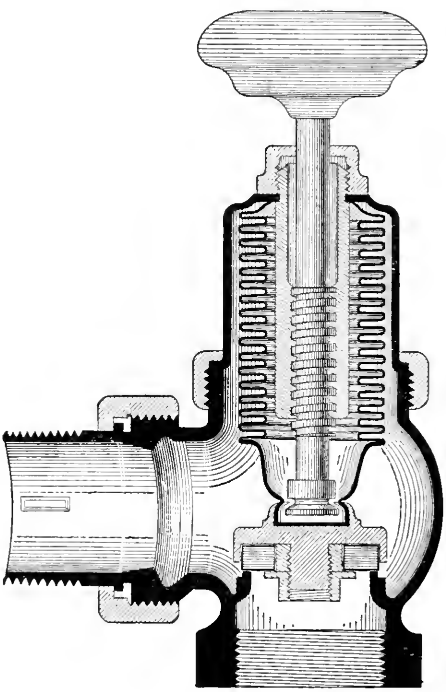


FIG. 33. SYLPHON PACKLESS RADIATOR VALVE.

The *Lavigne* packless radiator valve (Fig. 34) represents another type of packless valves, in which a permanent hard-rubber packing is used to prevent leakage at the spindle and a compression spring is used to hold the packing tightly against finished bearing rings. This valve is made with a very steep-pitched thread on the spindle so that it is *quick-opening,* moving through its full range with half a turn of the wheel or lever handle. The latter may be equipped with a pointer and indicator disc to show just how far open the valve is at any time, so that the amount of steam may be *graduated* to suit the heating requirements.

Graduated Steam Radiator Valves. These graduated fractional, or multi-port valves are *only employed on two-pipe* steam or, rarely, on water systems, and *cannot be used in one-pipe work.* They are useful only on steam, vapor, or vacuum systems in which a very sensitive pressure regulator is used, as they must be set or adjusted to supply each size of radiator at a definite supply pressure.

The *Detroit Multi-port* valve (Fig. 35) is a good example of this type and may be quickly altered to make the port any one of 18 openings, from 0.0215 up to 0.387 square inches. (Fig. 35*a*.)

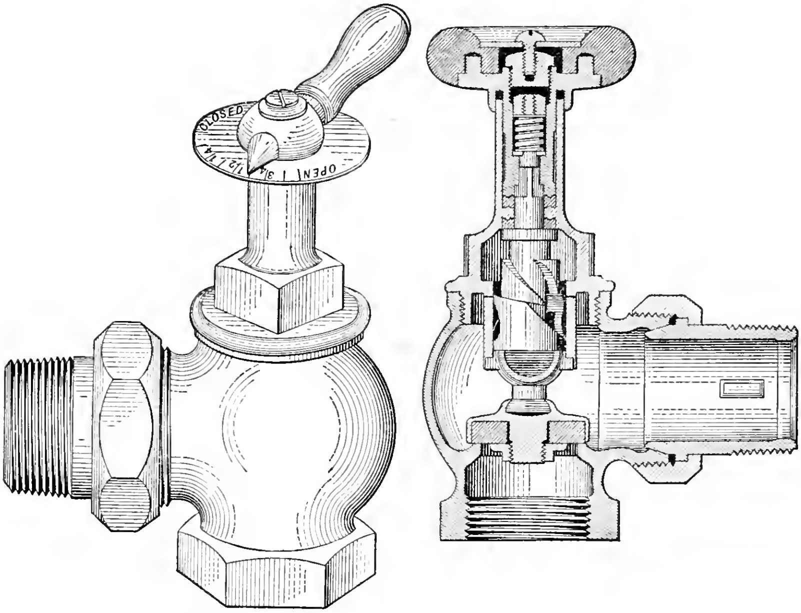


FIG. 34. LAVIGNE PACKLESS RADIATOR VALVE.

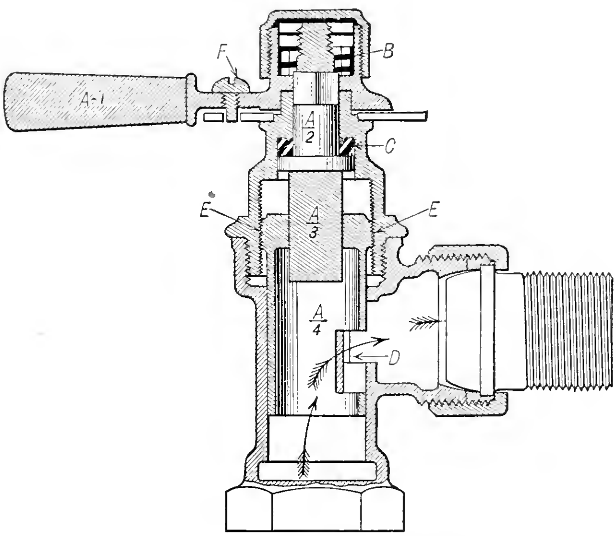


FIG. 35. DETROIT MULTI-PORT VALVE.

The port-regulating principle is illustrated by A-1, A-2, A-3, and A-4 (all A parts moving together). To change size of port, taking out the set-screw F permits the fitter to turn the handle A-1 completely around and the cylinder A-4 rises or falls on the thread E so that the port D is increased or decreased as required. (It is shown in the cut about one-half open.) The post A-3 works in a rectangular hole in top of the cylinder A-4—the latter rising or falling accord-

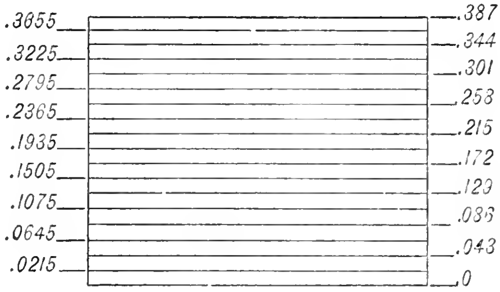


Fig. 35*a*. Showing the 18 Areas of Adjustable Ports in One Valve Running from 0.0215 to 0.387 Square Inch.

ing to the number of turns given to handle A-1. Each turn raises or lowers the post one thirty-second of an inch. Thus it is only a question of a few turns of the released handle A-1 to make the port just the size needed for any given radiator. With this valve *one* size (¾ inch) meets every one of 18 different supply conditions.

A series of tests were recently made to show the quantity of steam that would pass through a *Detroit Multi-port Vapor Valve* with shell wide open at pressures of two ounces, four ounces and eight ounces *at the valve*. These tests developed the fact that a ¾-inch multi-port valve with shell wide open passes 36.08 pounds of steam per hour on a two-ounce differential, and 51.63

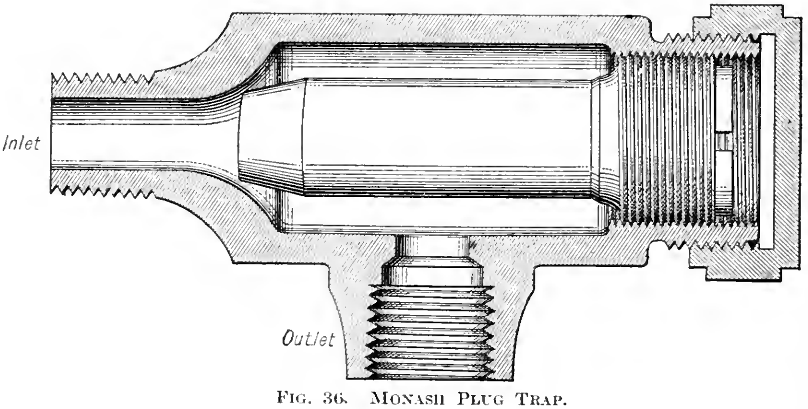


Fig. 36. Monash Plug Trap.

pounds of steam per hour at a four-ounce differential, and 72.35 pounds of steam per hour at an eight-ounce differential.

Assuming average condensation of one-quarter pound of steam per square foot of condensing surface per hour, these figures show that this valve with shell wide open will serve 144.32 square feet of condensing surface with a two-ounce differential, or 206.52 square feet at a four-ounce differential, or 494 square feet of surface at an eight-ounce differential.

With the valve in intermediate positions for *any given initial pressure* it is thus possible to

supply just enough steam to heat, say, ¼, ½, ¾ or ⅞ of the radiator as may be necessary to suit the weather conditions.

Automatic Air Valves and Traps. The use of automatic air valves on steam radiators, mains, trap-tanks, heaters, etc., has resulted in the development of an endless variety of these devices, of which only a few typical examples can be considered. See Chapter on "Direct Steam Heating."

The *automatic-expansion-post type* of steam air valve may be of either the solid or hollow post construction. The former (Fig. 36) has a solid composition plug which expands when surrounded by steam, thus closing the inlet and preventing its escape. If air or water enter the trap or valve, contraction of the plug takes place and the valve remains open

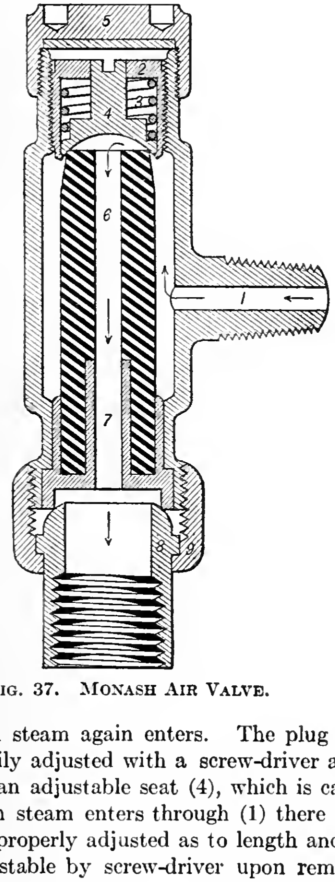


FIG. 37. MONASH AIR VALVE.

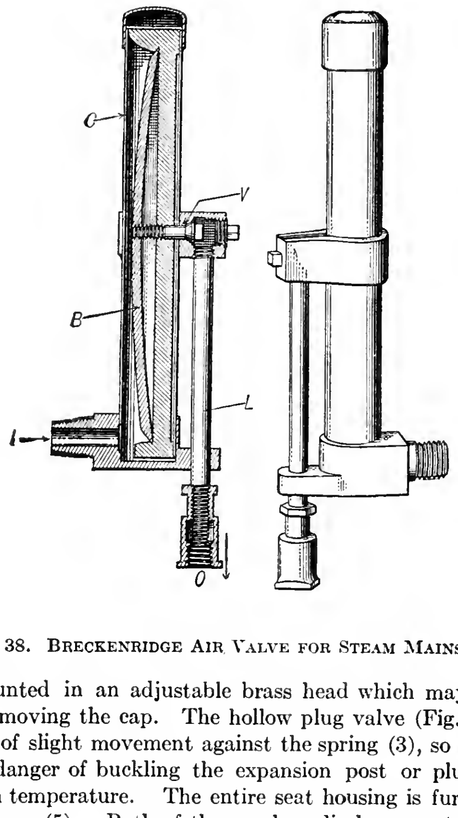


FIG. 38. BRECKENRIDGE AIR VALVE FOR STEAM MAINS.

until steam again enters. The plug is mounted in an adjustable brass head which may be readily adjusted with a screw-driver after removing the cap. The hollow plug valve (Fig. 37) has an adjustable seat (4), which is capable of slight movement against the spring (3), so that when steam enters through (1) there is no danger of buckling the expansion post or plug if not properly adjusted as to length and steam temperature. The entire seat housing is further adjustable by screw-driver upon removing cap (5). Both of these valves discharge water as well as air.

A modification of the expansion-post type of air valve is shown in the *Breckenridge* air valve (Fig. 38), in which a curved brass strip B in expanding pulls a small conical valve V against its seat, and prevents the escape of steam entering the chamber C, which contains the expansion element B. If air enters this same chamber the small valve V is moved to the right as the piece of spring brass contracts and the port is opened to the discharge line L connected to the small valve housing. Steam, air, and water enter at I, and air and water escape at O. The small valve may be readily adjusted by removing the protecting plug in its housing.

Especially perfected automatic steam radiator air valves for use in finished rooms have

been devised to discharge only air, and prevent the escape of steam and water, and are considered in the Chapter on "Direct Steam Heating."

Compression Cocks. Ordinary brass compression cocks (Figs. 39*a* and 39*b*) of ⅛″ or ¼″ size are generally found more satisfactory for use on hot-water systems than the automatic type of air valve. These cocks may be operated by nut and key, or preferably by a hardwood wheel, and, of course, can also be used for venting steam radiators and lines, where manual control of the air valves is satisfactory.

Finishes for Brass Valves. There are *five standard finishes* for all-brass radiator valves, as well as all-brass valves for general service. The methods of finishing these valves are: (1) rough body with finished trimmings, (2) finished all over, (3) rough body, with nickel-plated trimmings, (4) rough body, with finished trimmings and plated all over, and (5) finished and plated all over. For most interior work in finished rooms the finish specified is similar to number (4), while in unfinished rooms or basements number (1) is satisfactory. These are the two cheapest finishes to be had. Radiator control valves are usually fitted with hardwood wheels or handles, while valves on mains have iron wheels.

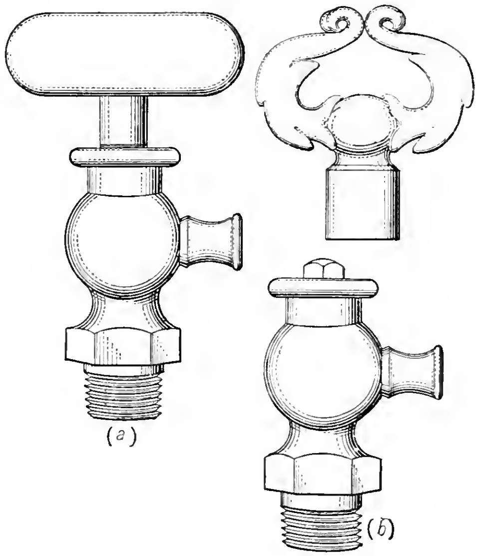


FIG. 39. COMPRESSION AIR COCKS.

COVERINGS

The proper *insulation* of air, water, and steam piping, valves, fittings, etc., must be carefully considered, not only from the standpoint of *heat loss*, which concerns steam and hot-water lines, but from the standpoint of *heat absorption*, which concerns cold-water piping as well as brine and ammonia lines. In fact, ordinary cold-water piping is often covered merely to prevent *sweating* in very hot weather.

Efficiency of Coverings. The heat loss to be overcome from steam and hot-water lines and the insulating efficiency of the various coverings depend upon the steam or water temperature and the porosity and thickness of the covering. No pipe covering is capable of preventing all the heat loss which would take place from bare uncovered pipe, but efficiencies as high as 86 per cent have been secured with commercial coverings, as shown in the following table and curves based on tests of various makes of covering. In every case the efficiency is understood to be the ratio of the heat-loss saving per sq. ft. to the heat loss per sq. ft. of bare pipe for the same internal and external temperature, rate of flow and external-air movement. Thus the efficiency of a covering is:

$$E = \frac{(H.L.B.) - (H.L.C.)}{H.L.B.} \times 100\%, \text{ in which}$$

$H.L.B.$ = heat loss in B.t.u. per sq. ft. per hr., bare pipe,

$H.L.C.$ = heat loss in B.t.u. per sq. ft. per hr., covered pipe.

The heat loss must be measured for the same temperature range and with all other conditions maintained identical in the two series of tests.

The heat loss from bare pipe ranges from 2 to 4 B.t.u. per sq. ft. per hour per degree difference in temperature between the steam or water flowing in the pipe and the external air. This *coefficient is not a constant* and is highest for small-pipe and large-temperature differences. In all

TABLE 26

APPROXIMATE EFFICIENCIES OF VARIOUS COVERINGS REFERRED TO BARE PIPES

Covering	Efficiency
Asbestocel	76.8
Gast's Air Cell	74.4
Asbestos Sponge Felt	85.0
Magnesia	83.5
Asbestos Navy Brand	82.0
Asbestos Sponge Hair	86.0
Asbestos Fire Felt	73.5
Cork	84.2–87.1

Based on one-inch covering and tests by *Paulding, Jacobus, Brill,* and others.

cases *still air* is supposed to surround the pipe, otherwise the loss *may far exceed* even 5 B.t.u. per sq. ft. per hour.

The *Babcock and Wilcox Co.* give the following table for heat losses from both bare and covered steam piping, with magnesia covering of varying thicknesses:

TABLE 27

HEAT LOSS FROM COVERED* AND UNCOVERED STEAM PIPES
CALCULATED FOR 160 POUNDS PRESSURE AND 60 DEGREES TEMPERATURE

Pipe, Inches	Thickness of Covering	½ inch	¾ inch	1 inch	1¼ inch	1½ inch	Bare
2	B.t.u. per lineal foot per hour	149	118	99	86	79	597
	B.t.u. per square foot per hour	240	190	161	138	127	959
	B.t.u. per square foot per hour per one degree difference in temperature	.770	.613	.519	.445	.410	3.198
4	B.t.u. per lineal foot per hour	247	193	160	139	123	1,085
	B.t.u. per square foot per hour	210	164	136	118	104	921
	B.t.u. per square foot per hour per one degree difference in temperature	.677	.592	.439	.381	.335	2.970
6	B.t.u. per lineal foot per hour	352	269	221	190	167	1,555
	B.t.u. per square foot per hour	203	155	127	110	96	897
	B.t.u. per square foot per hour per one degree difference in temperature	.655	.500	.410	.355	.310	2.89
8	B.t.u. per lineal foot per hour	443	337	276	235	207	1,994
	B.t.u. per square foot per hour	196	149	122	104	92	883
	B.t.u. per square foot per hour per one degree difference in temperature	.632	.481	.394	.335	.297	2.85
10	B.t.u. per lineal foot per hour	549	416	337	287	250	2,468
	B.t.u. per square foot per hour	195	148	120	102	89	877
	B.t.u. per square foot per hour per one degree difference in temperature	.629	.477	.387	.329	.287	2.83

* Covering—Magnesia, canvas covered.

NOTE.—For calculating radiation for pressure and temperature other than 160 pounds and 60 degrees, use B.t.u. figures for one degree difference. (Approximate only, as coefficient varies with the temperature range.)

Tests of Pipe Covering. The heat loss from bare and covered pipes has recently been determined in a series of tests conducted at the University of Wisconsin. The results of these tests have been reported at length in a paper presented at the annual meeting of the *A. S. M. E.*, December, 1915, by *L. B. McMillan.*

The tests were run on 5-inch diameter standard steel pipe, and a net length of 15 ft. was used for measuring the heat transmission of bare and covered pipe. The results are plotted

in the form of curves (Figs. 40 and 41), and the following statement applies to the test on uncovered pipe:

"The total loss curve in Fig. 40 is plotted directly from the data obtained during the test. The ordinate of any point is the total heat loss per hour, which is the equivalent of the electrical energy required to maintain the pipe at the given temperature, and the abscissa is the difference between pipe temperature and room temperature. On the same sheet is plotted a curve of heat losses per hour from the short pipe at various temperature differences; this curve is called "end correction." The difference of ordinates between the two curves at any value of temperature difference gives the net

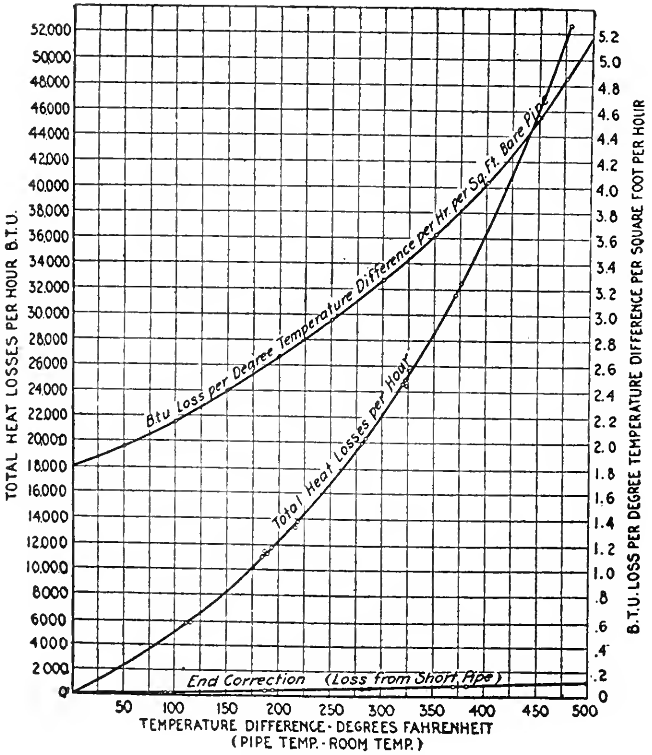


FIG. 40. TEST OF BARE PIPE.
(*L. B. McMillan.*)

heat loss per hour from the 15-ft. length of bare pipe. This net loss divided by the temperature difference and the area of test section (22.03 sq. ft.) gives the heat loss per degree temperature difference per square foot per hour.

"The curve of net heat losses per degree temperature difference per square foot per hour is shown in Fig. 40 to a much larger scale. This curve shows that the heat loss per degree temperature difference is far from being a constant at all temperatures, as has been assumed or implied by most former investigators."

All the coverings tested were bought in the open market, and *Mr. McMillan* gives the following description of the insulation as furnished ready for testing:

"The statements as to whether the covering was recommended for high- or low-pressure or superheated steam pipes were furnished by the manufacturers, and are not conclusions drawn from the tests.

The weight per foot in each case is the average weight per lineal foot of 5-in. covering, and the thickness given is the average thickness.

"I. *J-M 85 Per Cent Magnesia.* Molded sectional covering for high-pressure steam pipes. 85 per cent by weight of magnesium carbonate and the remainder principally asbestos fibre. Weight per foot, 2.92 lb., and thickness 1.08 in.

"II. *J-M Indented.* Layers of asbestos felt with indentations, about $1\frac{1}{4}$ in. in diameter and

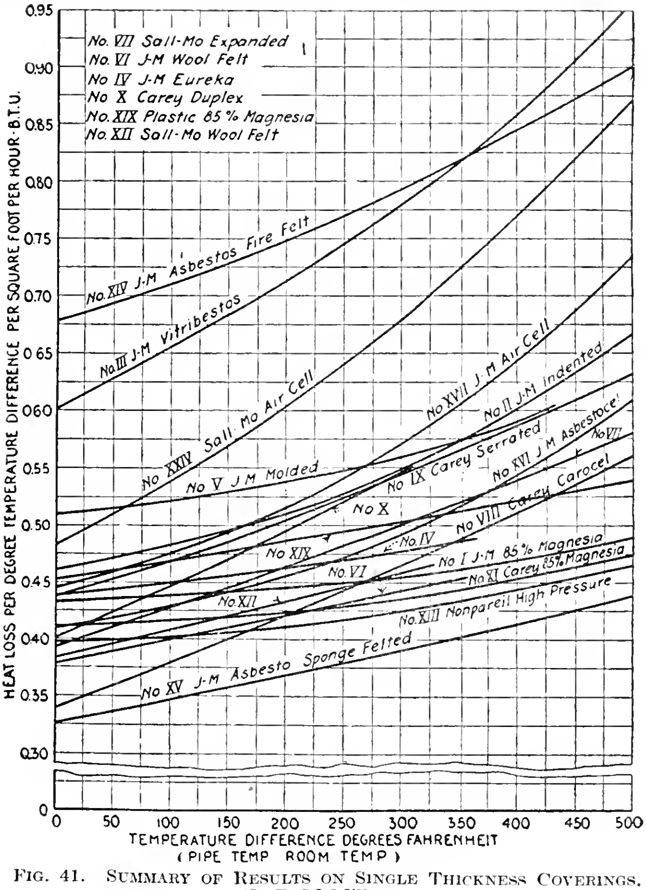


FIG. 41. SUMMARY OF RESULTS ON SINGLE THICKNESS COVERINGS. (*L. B. McMillan.*)

$\frac{1}{8}$ in. deep, spaced very close to each other in staggered rows. For pipes containing high-pressure steam. Weight per foot, 3.46 lb., and thickness 1.12 in.

"III. *J-M Vitribestos.* Asbestos air-cell covering made of alternate layers of smooth and corrugated vitrified asbestos sheets. Corrugations about $\frac{1}{4}$ in. deep and run lengthwise of the pipe. For use on high-pressure and superheated steam pipes, and for stack linings, etc. Weight per foot, 4.05 lb., and thickness 0.96 in.

"IV. *J-M Eureka.* For low-pressure steam and hot-water pipes. Made of $\frac{1}{4}$ in. of asbestos felt on the inside of the section and the balance of alternate layers of asbestos and wool felt. Weight, 4.60 lb. per ft., and thickness 1.04 in.

"V. *J-M Moulded Asbestos.* Moulded sectional covering for use on low and medium-pressure steam pipes. Made of asbestos fibre and other fireproof material. Weight per ft., 5.53 lb., and thickness is 1.25 in.

"VI. *J-M Wool Felt.* A sectional covering made of layers of wool felt with an interlining of two layers of asbestos paper. May be used on low-pressure steam and hot-water pipes. Weight per ft., 2.59 lb., and thickness 1.10 in.

"VII. *Sall-Mo Expanded.* A covering for use on high and low-pressure steam pipes. Made of eight layers of material, each consisting of a smooth and a corrugated piece of asbestos paper, the corrugations being so crushed down to form small longitudinal air spaces. Weight, 3.47 lb. per ft., and thickness 1.07 in.

"VIII. *Carey Carocel.* Composed of plain and corrugated asbestos paper firmly bound together. Corrugations are approximately 1/8 in. deep and run lengthwise of the pipe. For use on medium and low-pressure steam pipes. Weight, 3.06 lb. per ft., and thickness 0.99 in.

"IX. *Carey Serrated.* For high-pressure steam pipes. Composed of successive layers of heavy asbestos felt having closely spaced indentations. Weight, 5.66 lb. per ft., and thickness 1.00 in.

"X. *Carey Duplex.* For low-pressure steam and hot-water pipes. Alternate layers of plain wool felt and corrugated asbestos paper firmly bound together. Corrugations run lengthwise of the pipe and make air cells approximately 1/4 in. deep. Weight 1.79 lb. per ft. and 0.96 in. thick.

"XI. *Carey 85 Per Cent Magnesia.* For high-pressure steam and similar in composition to No. 1. Weight per foot, 2.74 lb. and thickness, 1.10 in.

"XII. *Sall-Mo Wool Felt.* Similar to No. VI except without interlining asbestos paper. For low-pressure steam and hot-water pipes. Weight per foot, 3.73 lb., and thickness 1.01 in.

"XIII. *Nonpareil High Pressure.* Moulded sectional covering consisting mainly of silica in the form of diatomaceous earth—the skeletons of microscopic organisms. For high-pressure and superheated steam pipes. Weighs 2.96 lb. per ft. and is 1.16 in. thick.

"XIV. *J-M Asbestos Fire Felt.* Asbestos fibre loosely felted together, forming a large number of small air spaces. For high-pressure and superheated steam pipes. Weight per ft., 3.75 lb., and thickness 0.99 in.

"XV. *J-M Asbestos Sponge Felted.* Made from a thin felt of asbestos fibre and finely ground sponge forming a very cellular fabric. Forty-one of these sheets per in. thickness; air spaces are formed between the sheets in addition to those in the felt itself. Specially recommended for high-pressure and superheated steam pipes. Weight per ft., 4.04 lb., and thickness 1.16 in.

"XVI. *J-M Asbestocel.* For medium-pressure steam and heating pipes. Alternate sheets of corrugated and plain asbestos paper forming air cells about 1/8 in. deep that run around the pipe Weight per ft., 1.94 lb., and thickness 1.10 in.

"XVII. *J-M Air Cell.* Corrugated and plain sheets of asbestos paper arranged alternately so as to form air cells about 1/4 in. deep running lengthwise of the pipe. For medium-pressure steam and heating pipes. Weight per ft., 1.55 lb., and thickness 1.00 in.

"XVIII. *1/2-In. J-M Plastic 85 Per Cent Magnesia.* For fittings, valves, irregular surfaces, boiler coverings, etc. Similar in composition to the sectional 85 per cent magnesia, but applied in the form of a cement or plaster. Thickness, 0.51 in. for the first test, and weight per ft. 1.51 lb.

"XIX. 1-*In. J-M Plastic 85 Per Cent Magnesia.* Thickness, 1.05 in.; weight per ft., 3.33 lb.

"XX. 1½-*In. J-M Plastic 85 Per Cent Magnesia.* Thickness, 1.48 in.; weight per ft., 5.23 lb.

"XXI. 2-*In. J-M Plastic 85 Per Cent Magnesia.* Thickness, 1.99 in.; weight per ft., 7.46 lb.

"XXII. 3-*In. J-M 85 Per Cent Magnesia.* The two inches of plastic covering of No. XXI and one standard thickness layer of sectional covering outside of that. Thickness, 3.24 in.; weight per ft., 11.67 lb.

"XXIII. ½-*In. Sall-Mo Air Cell.* Similar in composition and uses to No. XVII. Thickness, 0.51 in., and weight per ft. 0.99 lb.

"XXIV. 1-*In. Sall-Mo Air Cell.* Thickness, 0.95 in.; weight per ft., 1.57 lb.

"XXV. 2-*In. Sall-Mo Air Cell.* Thickness, 1.86 in.; weight per ft., 3.58 lb.

"XXVI. 3-*In. Air Cell.* Two inches of *Sall-Mo* and one inch of *J-M Air Cell.* Thickness, 3.04 in.; weight per ft., 6.66 lb."

The Economy of Using Pipe Covering. The saving to be effected by the use of pipe covering is readily calculated for any given condition as follows:

Example. Given a 3″ line 100 ft. in length, carrying steam at 80 lb. gage, and covered with one inch of 85 per cent magnesia having an insulating efficiency of 83.5 per cent as given by Table 26. The plant operates 10 hours per day for 300 days per year, and the average boiler-room temperature is 65° F. Coal costs $4.00 per ton.

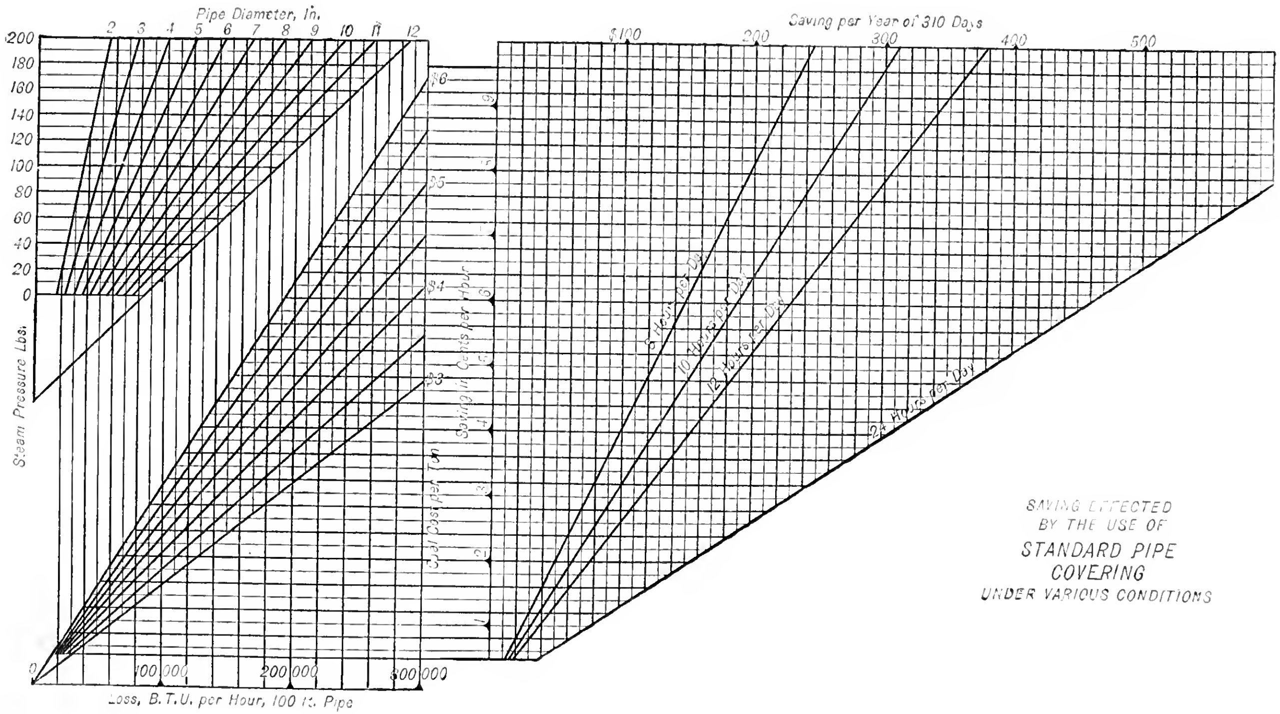

Pipe Diameter, In.
2 3 4 5 6 7 8 9 10 11 12
200 180 160 140 120 100 80 60 40 20 0
Steam Pressure Lbs.
0 100 000 200 000 300 000
Loss, B.T.U. per Hour, 100 ft. Pipe
$6 $5 $4 $3
Coal Cost per Ton
Saving in Cents per Hour
Saving per Year of 310 Days
$100 200 300 400 500
8 Hours per Day
10 Hours per Day
12 Hours per Day
24 Hours per Day
SAVING EFFECTED
BY THE USE OF
STANDARD PIPE
COVERING
UNDER VARIOUS CONDITIONS

FIG. 42.

The heat saved in this line with this covering in place is, assuming a low value for the coefficient at 80 lb., and taking $K = 2.65$ B.t.u. per hr., $H = 2.65 \times (324 - 65) \times 0.835 = 575$ B.t.u. per sq. ft. per hr. Hence, for a year of 300 days at 10 hours per day the heat loss is $= 575 \times 300 \times 10 = 1{,}725{,}000$ B.t.u., which is equal to $\frac{1{,}725{,}000}{13{,}500 \times 0.67} = 191.5$ lb. of coal saved per year. It is assumed a coal of 13,500 B.t.u. heat value per pound is burned with a boiler efficiency of 67 per cent.

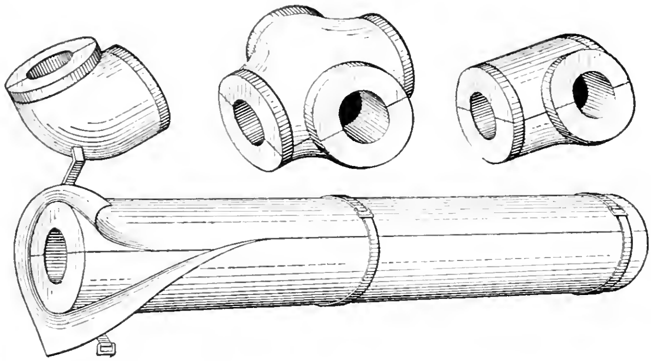


Fig. 43. Moulded 85% Magnesia Covering for Pipe and Fittings.

Since this coal costs \$4.00 per ton the saving per sq. ft. of pipe per year is $\frac{191.5 \times 400}{2000} = 38$ cents, and per 100 ft. of 3″ pipe $= 91.65 \times 0.38 = \$34.80$ per year.

The cost of magnesia covering at 65 per cent off the list is \$0.158 per lineal ft., or, in a year of 300 days, the saving in $\frac{100 \times 0.158}{34.80} \times 300 = 136$ days would pay for the covering.

The preceding chart (Fig. 42) by *H. C. Spaulding* indicates graphically the saving to be effected in heat units and dollars by standard covering, when applied to pipe of varying diameters,

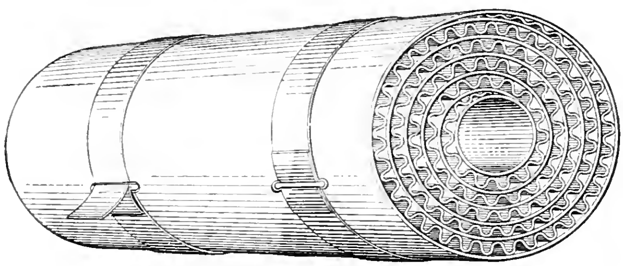


Fig. 44. Air Cell Covering—Plain or Vitrified Asbestos.

for a wide range in steam temperatures or pressures, with coal costing from three to six dollars per ton.

Commercial Pipe Coverings. The covering materials commercially available are usually moulded into cylindrical shape, in lengths of about 3′-0″ and of the proper diameter to fit all standard pipe sizes (Figs. 43 and 44). To facilitate their application, they are usually split lengthwise and are supplied with a suitable canvas jacket and the necessary fastening bands of about 30 B. and S. gauge solid sheet brass for holding the covering and its jacket in place.

The *loose covering material* in fibrous or granulated condition may also be secured in bulk for plastic application to various sorts of irregular surfaces. The more common coverings, such as magnesia, diatomaceous earth, cork, etc., are moulded into certain special forms (Fig. 43) to fit the principal valves and fittings used.

Insulating Materials. The materials used for insulating boilers, tanks, pipe lines, fittings, etc., range from hair, wool felt, silk, and asbestos fiber to granulated cork, powdered magnesium carbonate, and diatomaceous earth, as well as ordinary paper and asbestos paper, either plain, corrugated, or vitrified.

The *requirements* for a satisfactory insulating material are that it should be (1) a good non-conductor, (2) easily moulded and applied, (3) not liable to deterioration or attractive to vermin, (4) fireproof, (5) light in weight, and (6) capable of withstanding some abuse, and not affected by water or steam. Unfortunately all these characteristics are not possessed by any one covering, as, for example, hair or wool felt, which are the best possible insulators, are also subject to deterioration and vermin, and will not stand abuse.

Such coverings as asbestos, magnesia, diatomaceous earth, cork and vitrified asbestos air cells are probably most generally used and most nearly fulfill the above requirements.

Selection of Covering. The character of the service usually determines the *kind* of covering as well as the *thickness* to be applied. For *steam service* the thickness varies with the pressure of saturated steam and with the temperature of superheated steam, and in the best piping practice the following insulation requirements are observed.

Exposed radiating surfaces of all pipes, all high-pressure steam flanges, valve bodies and fittings, heaters and separators, should be covered with non-conducting material wherever such covering will improve plant economy. All main steam lines, engine and boiler branches, should be covered with 2 in. of 85 per cent carbonate of magnesia or the equivalent. Other lines may be covered with 1 in. of the same material. All covering should be sectional in form, and large surfaces should be covered with blocks, except where such material would be difficult to install, in which case plastic material should be used. In the case of flanges the covering should be tapered back from the flange in order that the bolts may be removed. Removable covers should be applied to the flanges.

All surfaces should be painted before the covering is applied. Canvas is ordinarily placed over the covering, and held in place by brass bands.

The *sectional moulded coverings*, such as 85 per cent magnesia (Fig. 43), are made in four thicknesses: (1) Standard, 1″; (2) Medium, 1½″; (3) Double standard, about 2″, and (4) Double medium, about 3″.

These same coverings are made up in blocks 3″x18″, 6″x36″, and range from ½″ to 4″ thick.

The *air cell sectional-moulded covering* (Fig. 44) is usually made in three thicknesses, of ½″, ¾″ and 1″ respectively, but is also supplied in heavier grade if required. This covering may be *vitrified* by dipping it in a vitreous bath, which, when properly treated, is capable of withstanding both water and great heat.

Moulded into flat or curved blocks this vitrified air-cell material is used for lining steel stacks and breechings and other surfaces subjected to excessive heat.

For *hot-water service* either one inch magnesia or earth covering is usually sufficient, applied as for steam.

For covering *ice* or *chilled water, brine, ammonia,* piping, etc., granulated cork, hair, or wool-felt covering is employed in thicknesses ranging from 1″ to 2″, depending on the temperature differential. (See "Refrigeration," Volume II.) Four ranges are recognized for cork: (1) Standard Brine Covering, 0° to 25° F., (2) Special Thick-Brine Covering, below 0° F., (3) Ice-Water Covering, 25° to 45° F., and (4) Cold-Water Covering, above 45° F.

The latest practice in specifying insulation material is to call for a *guaranteed insulating efficiency* or maximum permissible heat loss from or to the covered lines, and leave the question of the material, the thickness, and the method of application of the covering to the contractor, who must use such covering as will satisfy the guaranteed requirements.

STEAM-PLANT ACCESSORIES

A great variety of *secondary equipment* is required in large steam plants, and even in small plants more or less of this apparatus is essential for the proper installation and operation of the system.

Pipe Hangers. Suitable pipe hangers, supports, brackets, rollers, and guides must be used in the erection of all water-steam and air piping, and although this equipment is standard to some extent, special conditions are constantly arising which require new designs or modifications of old ones as shown in Figs. 45 and 46, where a variety of pipe hangers and supports are illus-

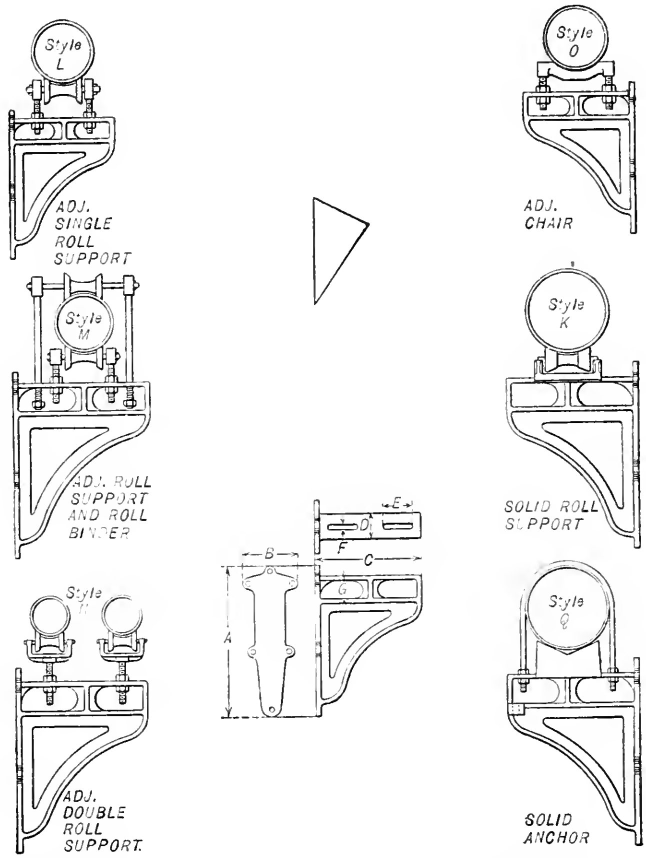


FIG. 45. PIPE SUPPORTS, BRACKETS, ROLLS, CHAIRS, ANCHORS, ETC.
Suitable for Pipe Lines from 5 to 30 inch.
(*Crane Co.*)

TABLE 28
DIMENSIONS OF BRACKETS

Size	Safe Load	Size of Pipe Will Support	A	B	C	D	E	F	G
No. 11	1 Ton	5 to 8 in.	34	12	25	6	8½	1¼	5½
No. 12	2 Tons	9 to 14 in.	40	14	30	6	9	1⅜	6
No. 13	3 Tons	15 to 18 in.	45	16	34	6	9¼	1½	6½
No. 14	4 Tons	20 to 24 in.	51½	19	40	6	11⅛	1⅝	7
No. 15	Special	20 to 30 in.	64	19	44⅛	6	12¼	1⅝	7

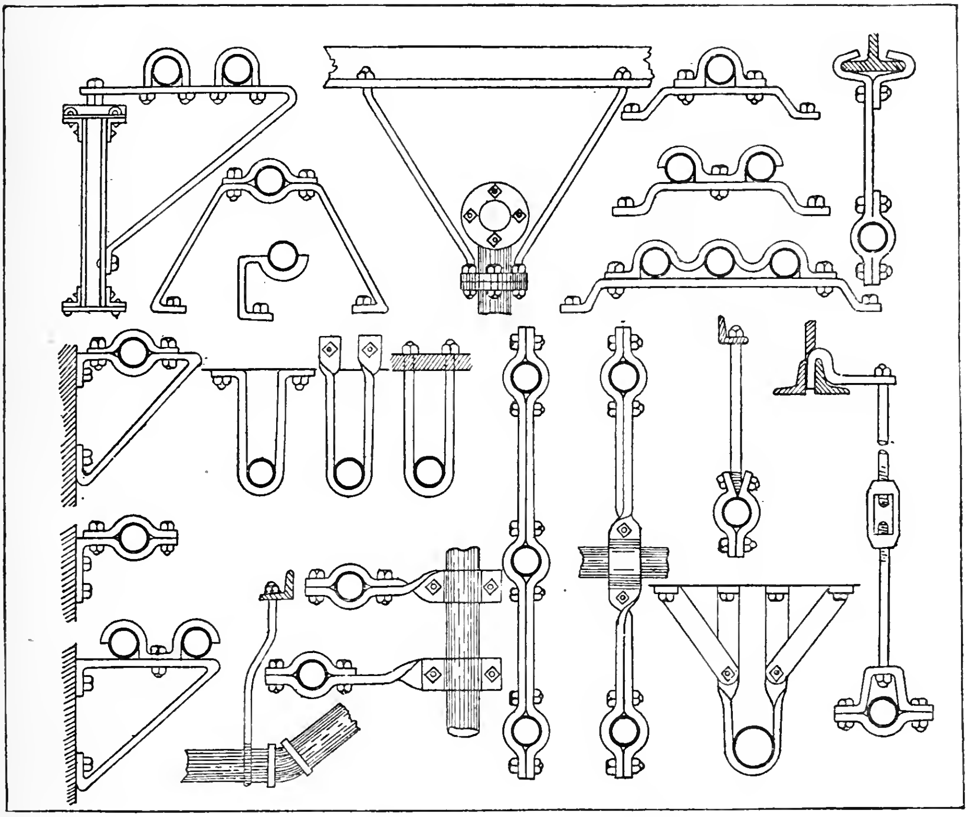


FIG. 46. VARIOUS FORMS OF PIPE HANGERS.

SPRINKLER PIPE HANGERS

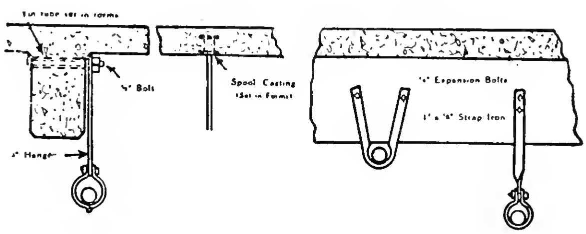


FIG. 46*a*.

DETAILS FOR ATTACHING HEAT PIPES

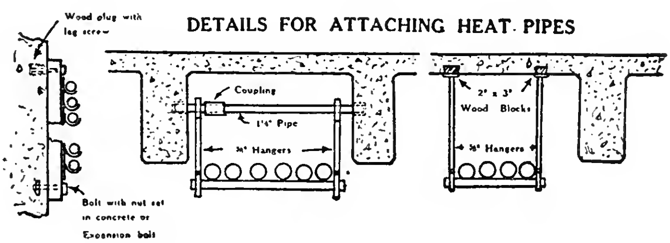


FIG. 46*b*.

trated. Their application depends entirely on the special conditions to be met, and the impossibility of attempting to standardize such equipment is self-evident.

Certain more or less *standard forms of hangers* are made, however, by most manufacturers, and an approved type of adjustable cast-iron hanger is shown in Fig. 47 for hanging pipe from

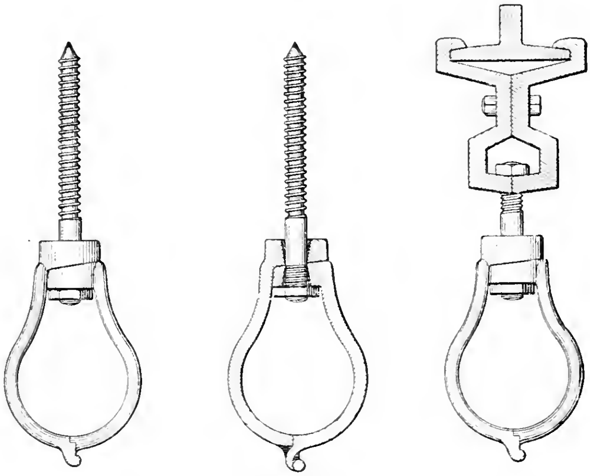


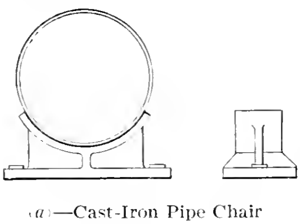
FIG. 47. PITTSBURGH PIPE HANGER.

wooden or steel beams. The pipe collars or yokes of these hangers are made in sizes for ½″ to 8″ pipe, and the beam clamps are made for beams with from 3″ to 8″ flanges.

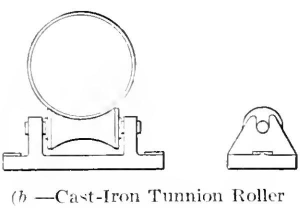
All pipe hangers and supports should be readily *adjustable* (Figs. 46 and 47) and provide for *reasonable movement* of the piping due to expansion and contraction. In case the supporting medium to which the hanger is attached is subject to unusual vibrations not found in all parts of the system, the hanger should be provided with *shock-absorbing springs* to overcome such vibration as far as possible.

The *design and proportions* of pipe clamps and hangers for pipes from 4″ to 17″ in diameter is shown in Tables 29, 30, and 31, following, which were compiled by *L. S. Richardson* and reported in "Machinery." The supports or hangers are ordinarily placed from 10 to 15 ft. on centers depending on pipe size, and must not only be near enough together not to exceed the allowable fiber stress in the rod, but also to keep the pipe from sagging, which in the smallest sizes of less than 4″ diameter may require a spacing of not more than 8 ft. between hangers.

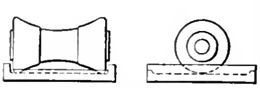
Pipe Rollers and Supports. In addition to the brackets and hangers already shown, a standard assortment of pipe chairs, bearings and rollers may be obtained for supporting pipe in trenches, or for attachment to brackets or hangers. (Figs. 55 and 58.)



(*a*)—Cast-Iron Pipe Chair or Support.



(*b*)—Cast-Iron Tunnion Roller Pipe Support.



(*c*)—Cast-Iron Plate Roller Pipe Support.

FIG. 55.

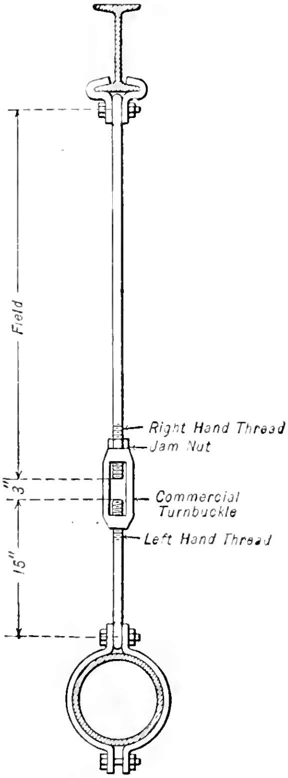


Fig. 48. Vertical Type.

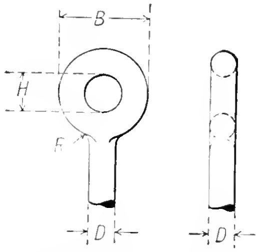


Fig. 49. Detail of Eye Bolt End.

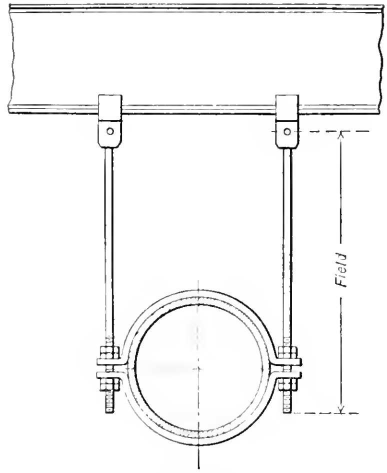


Fig. 50. Horizontal Type.

Size of Rod (D)	B	H	R
$\frac{3}{4}$	$2\frac{1}{2}$	1	1
$\frac{7}{8}$	$2\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$
1	$3\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$

All Dimensions in Inches

The size of the rod is determined by the size of the pipe. See Tables 30 and 31.

The size of the beam clamp is determined by the size of the beam and the size of the rod. See Table 29.

For the vertical type make the lower rod 15 inches long with left-hand thread. These can be made up in lots and be carried in stock.

Upper rod to have right-hand thread and one jam nut. Determine length of rod in field.

Use commercial turnbuckles.

Figure clearance between upper and lower rods as about 3 inches.

Rods are not upset at ends. Make threads 6 inches long. Horizontal type rods to have 4 nuts each, right-hand thread.

Determine length of rod in field.

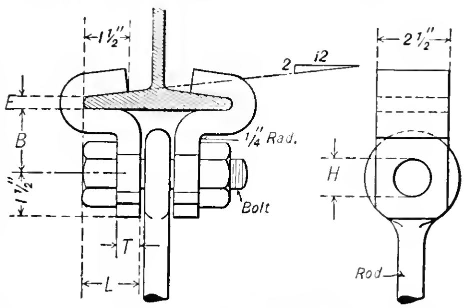


Fig. 51. Detail of I-Beam Clamp (Open Type.)

TABLE 29

I-BEAM CLAMPS

All Dimensions in Inches. All Loads in Pounds.

Loads Based on Fiber Stress of 12,000 Pounds for Wrought Iron	
$\frac{3}{4}$ rod	3500
$\frac{7}{8}$ rod	5000
1 rod	6500

Size of Rod	T	B	H	Size of Bolt	Size of Rod		Size of Beam 8	9	10	12	15	18	20	24
$\frac{3}{4}$	$\frac{5}{8}$	$1\frac{3}{4}$	1	$\frac{7}{8}$ x $3\frac{3}{4}$	All Rods	E	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$
$\frac{7}{8}$	$\frac{3}{4}$	2	$1\frac{1}{8}$	1 x $4\frac{1}{4}$	$\frac{3}{4}$	L	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{16}$	$2\frac{5}{16}$	$2\frac{9}{16}$	$2\frac{11}{16}$	$3\frac{1}{16}$
1	$\frac{7}{8}$	$2\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{8}$ x $4\frac{3}{4}$	$\frac{7}{8}$	L	$1\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{13}{16}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{5}{8}$	3
					1	L	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{7}{16}$	$2\frac{9}{16}$	$2\frac{15}{16}$

TABLE 29. (*Continued*)

Size of Rod	*C*	*Z*	*F*	*W*	*K*
3/4	7/8	1	1/2	2	1 5/8
7/8	1	1 1/8	5/8	2 1/4	1 7/8
1	1 1/8	1 1/4	3/4	2 1/2	2 1/8

Size of Bolt	Size of Beam	*A*	*D*
7/8 x 3 1/4	3	2 3/4	3 1/8
1 x 3 3/4	4	3 1/8	4 1/8
1 1/8 x 4 1/4	5	3 1/2	5 1/8
	6	3 3/4	6 1/8
	7	4	7 1/8

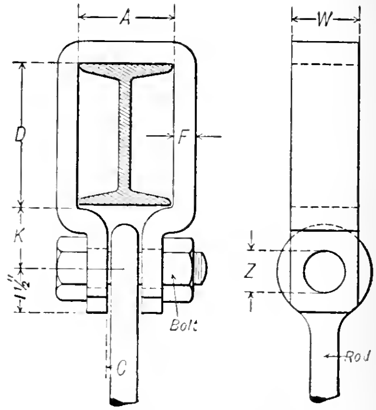


FIG. 52. DETAIL OF I-BEAM CLAMP. (LOOP TYPE.)

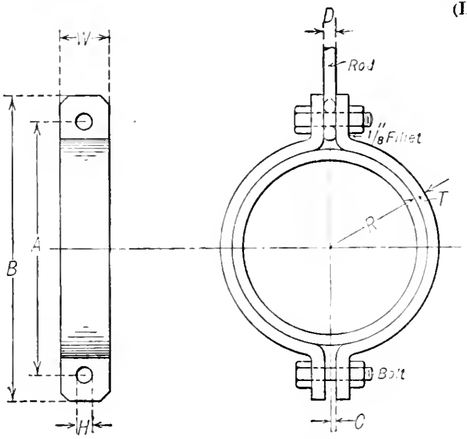


FIG. 53. VERTICAL TYPE. (See Table 30.)

TABLE 30
PIPE CLAMPS AND HANGERS
All Dimensions in Inches

FOR ALL CLAMPS							FOR PIPE CLAMPS				FOR FITTING CLAMPS			
Size of Pipe	*C*	*D*	*H*	*T*	*W*	Size of Bolt	Size of Pipe	*A*	*B*	*R*	Size of Pipe	*A*	*B*	*R*
4	1/2	3/4	1	3/8	1 3/4	7/8 x 3	4	7 3/8	9 7/8	2 1/4	4	8 5/8	11 1/8	2 7/8
4 1/2	1/2	3/4	1	3/8	1 3/4	7/8 x 3	4 1/2	7 7/8	10 3/8	2 1/2	4 1/2	9 1/8	11 5/8	3 1/8
5	1/2	3/4	1	3/8	2	7/8 x 3	5	8 1/2	11	2 13/16	5	9 5/8	12 1/8	3 3/8
6	1/2	3/4	1	3/8	2	7/8 x 3	6	9 1/2	12	3 5/16	6	10 3/4	13 1/4	3 15/16
7	1/2	3/4	1	3/8	2	7/8 x 3	7	10 1/2	13	3 13/16	7	11 3/4	14 1/4	4 7/16
8	9/16	7/8	1 1/8	7/16	2	1 x 3 1/2	8	11 7/8	14 5/8	4 5/16	8	13 1/4	16	5
9	9/16	7/8	1 1/8	7/16	2 1/4	1 x 3 1/2	9	12 7/8	15 5/8	4 13/16	9	14 1/4	17	5 1/2
10	9/16	7/8	1 1/8	7/13	2 1/4	1 x 3 1/2	10	14	16 3/4	5 3/8	10	15 3/8	18 1/8	6 1/16
12	5/8	1	1 1/4	1/2	2 1/2	1 1/8 x 4	12	16 3/8	19 3/8	6 3/8	12	18 1/8	21 1/8	7 1/4
14 / 15 O. D. .	5/8	1	1 1/4	1/2	2 1/2	1 1/8 x 4	14 / 15 O. D.	18 5/8	21 5/8	7 1/2	14 / 15 O. D.	20 3/8	23 3/8	8 3/8
15 / 16 O. D. .	5/8	1	1 1/4	1/2	2 1/2	1 1/8 x 4	15 / 16 O. D.	19 5/8	22 5/8	8	15 / 16 O. D.	21 1/2	24 1/2	8 15/16
16 / 17 O. D. .	5/8	1	1 1/4	1/2	2 1/2	1 1/8 x 4	16 / 17 O. D.	20 5/8	23 5/8	8 1/2	16 / 17 O. D.	22 5/8	25 5/8	9 1/2

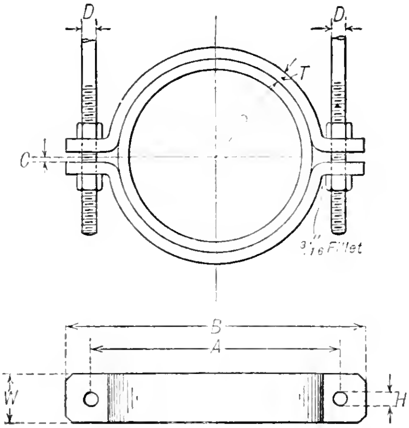


FIG. 54. HORIZONTAL TYPE.

(See Table 31.)

TABLE 31

PIPE CLAMPS AND HANGERS

All Dimensions in Inches

For All Clamps						For Pipe Clamps				For Fitting Clamps			
Size of Pipe	*C*	*D*	*H*	*T*	*W*	Size of Pipe	*A*	*B*	*R*	Size of Pipe	*A*	*B*	*R*
4..........	3/16	3/4	7/8	3/4	1 3/4	4..........	8	10 1/4	2 1/4	4..........	9 1/4	11 3/4	2 7/8
4 1/2..........	3/16	3/4	7/8	3/4	1 3/4	4 1/2..........	8 1/2	10 3/4	2 1/2	4 1/2..........	9 3/4	12 1/4	3 1/8
5..........	3/16	3/4	7/8	3/4	2	5..........	9 1/8	11 3/8	2 13/16	5..........	10 1/4	12 3/4	3 3/8
6..........	1/4	3/4	7/8	3/4	2	6..........	10 1/8	12 3/8	3 5/16	6..........	11 3/8	13 7/8	3 15/16
7..........	1/4	3/4	7/8	3/4	2	7..........	11 1/8	13 3/8	3 13/16	7..........	12 3/8	14 7/8	4 7/16
8..........	1/4	3/4	7/8	3/4	2	8..........	12 1/8	14 3/8	4 5/16	8..........	13 1/2	16	5
9..........	1/4	3/4	7/8	3/4	2 1/4	9..........	13 1/8	15 3/8	4 13/16	9..........	14 1/2	17	5 1/2
10..........	1/4	3/4	7/8	3/4	2 1/4	10..........	14 1/4	16 1/2	5 3/8	10..........	15 5/8	18 1/8	6 1/16
12..........	5/16	7/8	1	1	2 1/2	12..........	17	19 1/2	6 3/8	12..........	18 3/4	21 1/2	7 1/4
14 / 15 O. D....	5/16	7/8	1	1	2 1/2	14 / 15 O. D...	19 1/4	21 3/4	7 1/2	14 / 15 O. D...	21	23 3/4	8 3/8
15 / 16 O. D....	5/16	7/8	1	1	2 1/2	15 / 16 O. D...	20 1/4	22 3/4	8	15 / 16 O. D...	22 1/8	24 7/8	8 15/16
16 / 17 O. D....	5/16	7/8	1	1	2 1/2	16 / 17 O. D...	21 1/4	23 3/4	8 1/2	16 / 17 O. D...	23 1/4	26	9 1/2

In many cases it is necessary to support piping at some distance from the floor on columns or frames resting on the floor with a yoke at top of the adjustable column (Fig. 56) or with an adjustable hanger (Fig. 57), in case of a frame, which always makes a more stable support.

High Pressure Steam Traps. The use of steam traps for automatically draining the condensation from steam lines at varying pressures is very generally practised in all steam plants except those in which the water of condensation, as in low pressure direct heating systems, returns to the boiler by gravity. These devices, of which an approved form is shown in Fig. 59, are so arranged, with outlet valve under the automatic control of a ball float or an open bucket float, that when the receiving chamber of the trap fills with water of condensation, as indicated by the gauge glass, the float automatically opens the discharge valve, and if the steam pressure

at the inlet is greater than the total head at the outlet the water is driven out until the falling float again closes the discharge valve. This latter valve must always be protected by a water seal of 2 or 3 inches to prevent any possibility of steam blowing through same. A suitable drain or blow-off and self-contained by-pass should be provided, and also an air valve, for venting trap of any accumulation of air which may occur. The *Anderson* model "D" steam trap (Fig. 59) possesses all these features, and is readily inspected in case of trouble without having to break the steam joints or pipe connections to the trap.

Low pressure steam traps are described in detail in the Chapter on "Direct Steam Heating," in Volume I.

Separating or *non-return traps* like the above will not ordinarily return water to the boiler when the steam supply for the trap is taken from this same boiler. A combination of two traps is necessary for this service and a *tilting type of return trap* is generally used, placed from 3 to 4 ft. above the boiler water-line, into which the separating traps can discharge. This type of trap is also discussed in detail in the Chapter on "Direct Steam Heating," in Volume I.

Steam and Oil Separators and Exhaust Heads. The removal of fine particles of water and oil, which are often "entrained" or caught up by and carried along with the current of steam, is accomplished by the use of *steam and oil separators* (Fig. 60) and exhaust heads (Fig. 61). This apparatus is placed in the line di-

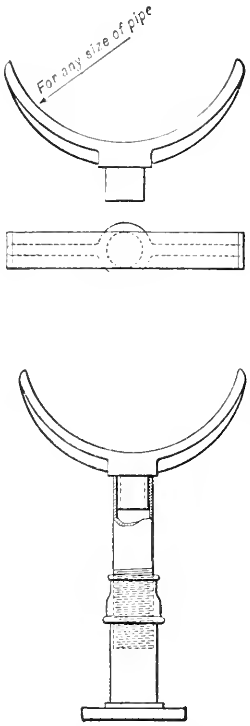


Fig. 56. A Column Pipe Support with Yoke.

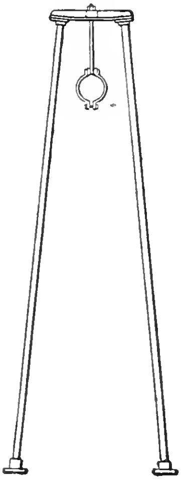


Fig. 57. A Frame Pipe Support with Flexible Pipe Hanger.

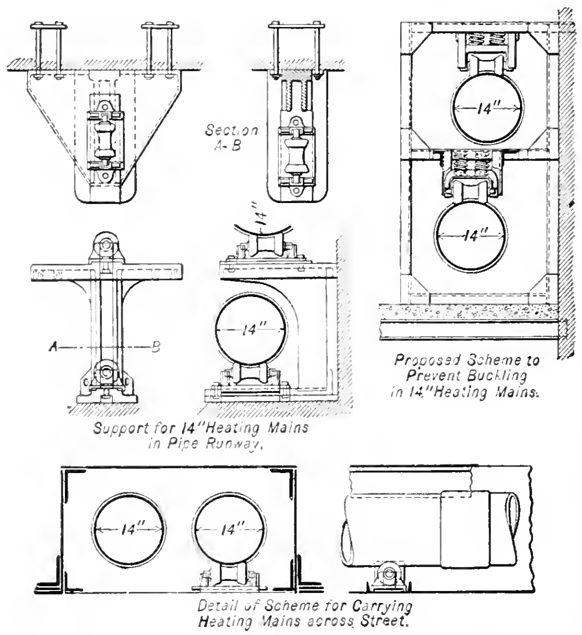


Fig. 58. Details of Roller Supports.

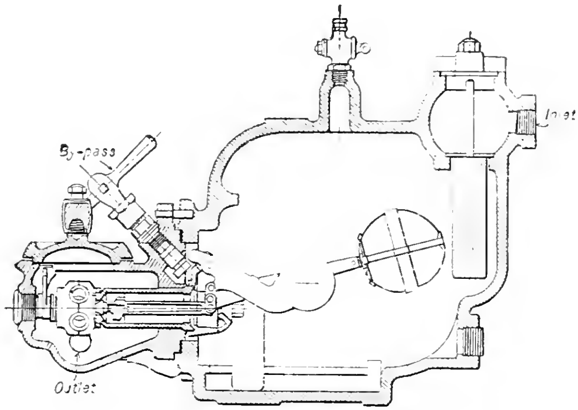


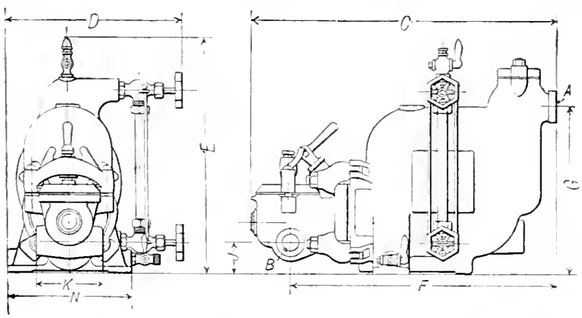
FIG. 59. ANDERSON MODEL "D" STEAM TRAP.
(See Tables 32 and 33.)

TABLE 32
SIZES AND CAPACITIES OF THE ANDERSON MODEL "D" STEAM TRAP

Size number of trap	1	2	3	4	5	6	7
Size of pipe connection, in inches	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2
Maximum discharge of condensation, per hour in pounds	1,500	2,400	4,000	5,600	8,000	12,000	24,000
Greatest number of square feet of surface that should be applied	1,000	1,600	2,600	4,700	7,000	10,000	20,000
Greatest number of lineal feet of 1-inch pipe surface that should be applied	3,000	5,000	8,000	14,000	20,000	30,000	60,000
Net weight of complete trap, in pounds	81	92	150	166	268	321	525
Shipping weight, in pounds (boxed)	110	114	175	200	335	394	620

NOTE.—Standard steam traps are suitable for pressures from 150 pounds down to 30 pounds. Low pressure steam traps are suitable for pressures from 30 pounds down. Always state maximum steam pressure at the trap.

TABLE 33
DIMENSIONS IN INCHES OF THE ANDERSON MODEL "D" STEAM TRAP



Size Number	1	2	3	4	5	6	7
A	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2
B	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2
C	19 1/4	20 3/8	24 1/4	25 3/4	30 3/8	32 1/2	39
D	11 1/4	11 1/4	12 5/8	13 1/4	14 5/8	15 1/2	17 1/4
E	15 1/2	16 1/8	18	18 3/4	22	23 3/4	27 5/8
F	16 7/8	18 1/4	21	22 5/8	26 3/8	28 1/2	33 1/4
G	10 3/4	11 5/8	13 1/2	14 5/16	17	18 5/8	22
J	2	2 1/16	1 13/16	2	2 1/8	2 1/8	3 1/8
K	4 1/4	4 1/4	4 7/8	4 7/8	6 1/2	6 5/8	8 1/8
N	7 3/4	7 3/4	9 1/8	9 1/2	11 1/8	12	14

rectly in the path of the steam, and by suitable baffles, which intercept the rapidly moving particles of liquid, collect or separate the water and oil from the steam. The greater momentum of these particles causes them either to drive ahead against the baffles or to be thrown

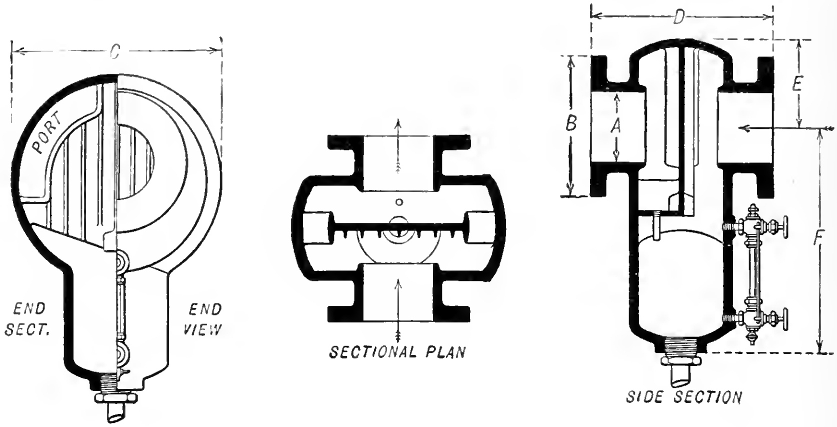


Fig. 60. Cochrane Horizontal Type Oil and Steam. (See Table 34.)

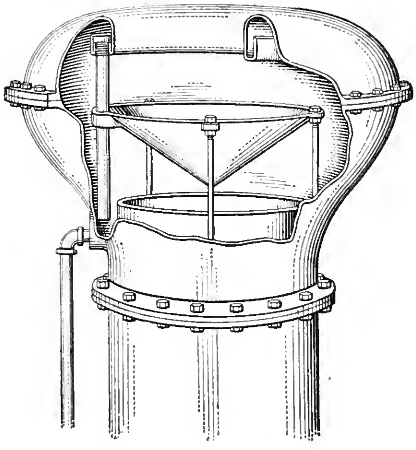


Fig. 61. Exhaust Head. Cast Iron.

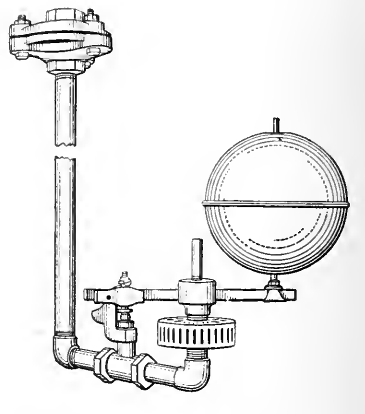


Fig. 62. Climax Automatic Cellar Drainer.

to the outside of the separating chamber by centrifugal force when the direction of the steam current is suddenly altered. In this way separation is effected, and the water and oil drained away by suitable drips.

The steam and oil separators are usually of cast iron, and are generally built in sizes from 3″ to 12″ for low pressure, standard, and extra-heavy service, with corresponding flanges which may be readily bolted to flanges or fittings of the *American Standard* schedule for standard or extra-heavy duty. Sizes down to 1½″ and up to 36″ diameter may be obtained for special service.

Exhaust heads (Fig. 61) are preferably made of cast iron, although a number of designs in

sheet metal are made. The use of these heads is most essential at the top of all atmospheric exhaust lines if " dry " steam is to be discharged. If the *oil* and *water* entrained in the exhaust

TABLE 34

DIMENSIONS OF COCHRANE STEAM AND OIL SEPARATORS

For Non-Condensing Systems. Any Working Pressure 50 Lb. per Sq. Inch or Under. Sizes, 3 to 12 Inches, Inclusive.

All Dimensions Given in Inches

Size of Pipe (I. D.)	Approx. Weights		Principal Dimensions of Standard Sizes						
	Stripped	Complete	*A*	*B*	*C*	*D*	*E*	*F*	Drip
3	118	150	3	7 ½	10 ½	11	4 ⅜	17	¾
3 ½	138	175	3 ½	8 ½	11 ½	11 ¾	4 ¾	17	¾
4	155	200	4	9	12 ¾	12 ¼	5 ⅜	17	¾
4 ½	180	225	4 ½	9 ¼	13 ¾	12 ¾	5 ¾	18	1
5	199	250	5	10	15	13 ¼	6 ⅜	19	1
6	234	300	6	11	17 ¼	14 ½	7 ⅜	20 ½	1
7	356	430	7	12 ½	19 ½	15 ½	8 ⅜	22	1 ¼
8	424	510	8	13 ½	21 ¾	16 ¾	9 ⅜	23 ¼	1 ¼
10	731	850	10	16	26 ½	18 ¾	11 ½	26	1 ½
12	1,113	1,280	12	19	31	21	13 ½	29	1 ½

steam are not removed the destruction and contamination of exposed roofs and walls are almost certain to result from the artificial rain developed. These heads are built in all sizes up to 36″ pipe diameter.

Water Ejectors or Drainers. The use of *automatic ejectors* and *cellar drainers* for removing accumulated water from low points, such as *cellars, wheel pits, furnace* and *boiler pits, foundations,* etc., when the lift is small is commonly practised if no sewer or drain is available at a lower level. These drainers may be operated by either steam or water at pressures of 15 lb. gage or more, beginning with a minimum lift of 5 to 6 ft. and increasing to 12 ft. at 80 lb. The *Climax* drainer (Fig. 62), *J. B. Clow and Sons,* has the following capacities at varying pressures when using water, and has the working parts made of brass to prevent corrosion. The apparatus is so installed that with an accumulation of 8″ of water in the sump or pit the float will open the supply cock and discharge the dead water accumulated.

TABLE 35

CLIMAX DRAINERS AND CAPACITIES

(See Fig. 62)

No.	Pressure, Lb.	Lift, Feet	Capacity Gallons per Hour	Supply Pipe	Discharge Pipe
1	15 to 80	6 to 12	50 to 250	½″	1″
2	15 to 80	6 to 12	100 to 400	¾	1 ¼
3	15 to 80	6 to 12	150 to 600	1	1 ½
4	15 to 80	6 to 12	200 to 800	1 ¼	2
5	15 to 80	6 to 12	275 to 1,000	1 ½	2 ½
6	15 to 80	6 to 12	350 to 1,200	2	3

Sample Specification. Furnish and install one *Climax* or equal cellar drainer, of a capacity of not less than (variable) gallons per hour. This ejector will be installed in a suitable sump pit made of concrete of dimensions to accommodate the ejector. All necessary connections to the water-service piping and waste connections from the drainer must be made as directed by the superintendent.

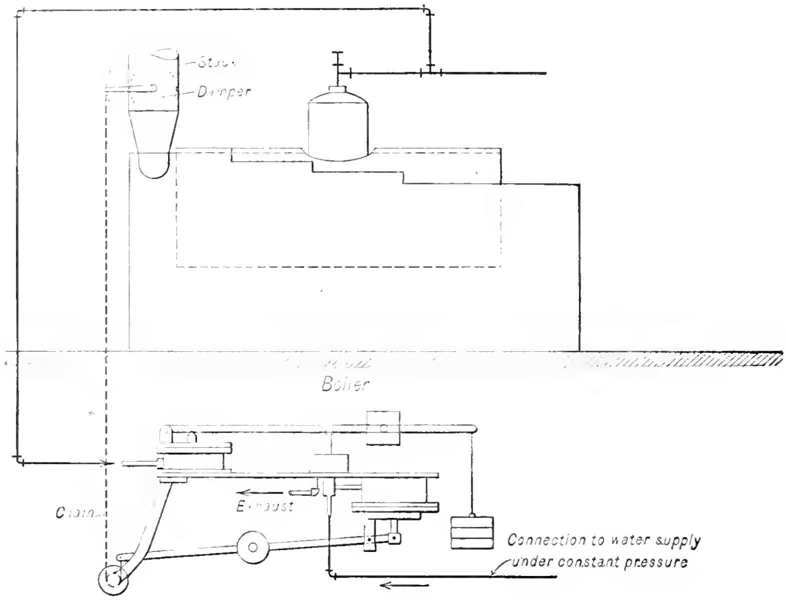


FIG. 63. HYDRAULIC DAMPER REGULATOR.

Damper Regulators for Power Boilers. The automatic control of the steam pressure in power boiler service is readily accomplished by varying the intensity of the draft through the fuel bed. In order to do this it is only necessary to connect an hydraulically operated damper regulator (Fig. 63) to the main boiler damper in the smoke breeching. This regulator, under the direct influence of the steam pressure within the boiler, moves the damper so as to regulate the draft in accordance with the demand for steam, and at the same time maintain the pressure practically constant.

CHAPTER XXI

PREPARATION OF PLANS AND SPECIFICATIONS AND COST OF HEATING AND VENTILATING EQUIPMENT

PREPARATION OF PLANS AND SPECIFICATIONS

Plans. It is generally customary in the best engineering practice to prepare complete plans of the heating and ventilating or mechanical equipment, showing the location and space required for the more important apparatus and fixtures as well as all runs of piping, including valves, special fittings, and the principal accessories. This may require the preparation of complete or partial *sectional elevations* and detailed drawings of the more complicated or unusual portions of the proposed installation.

These plans are usually drawn to *scale*, using either $\frac{1}{8}'' $ or $\frac{1}{4}'' = 1' - 0''$, depending on the extent of the building and the practical limitations involved both in making and using very large drawings. The details are ordinarily shown on a larger scale, usually, $\frac{3}{4}'' = 1' - 0''$, but here again practical limitations as to size of sheet must be taken into consideration in selecting the scale to be used.

It is, of course, always desirable to have *separate drawings for the mechanical equipment* if possible in order to avoid any possibility of confusion resulting due to crowding this work upon the same sheet with the architectural plans. In this way the mechanical equipment drawings are readily kept separate for the use both of the bidders and the supervising engineer. Furthermore, since only the more important architectural outlines need be shown on these sheets, it is possible to show all features of the equipment with greater detail and emphasis than would otherwise be possible. In fact, in many of the best layouts the architectural outlines are actually subdued by using a diluted ink in preparing the tracings.

The structural and architectural requirements must be kept constantly in mind in laying out the equipment, and the greatest care exercised to prevent *conflict* or *interference* with same. In fact, this is one of the prime reasons for laying this equipment out to scale in advance, and the engineer should take great pains to furnish the architectural designers with the exact, or at least, the minimum clearance dimensions required for the proper installation of all parts of the equipment. This applies not only to the floor space and minimum ceiling heights of engine and boiler rooms and elevator shafts, but to the space for running pipes and ducts. Structural beams and columns cannot be moved or cut once they are in place, or even after the plans have been sent to bidders. It is, therefore, most essential that the engineer should confer with the architect as soon as the design work is started and the general character and proportions of the building have been decided upon. Special attention should be devoted to the possible *interference of the piping systems* with any part of the steel work, since such a conflict if not anticipated in time may result in an expensive extra after erection has started.

In this connection, it is very necessary to keep always in mind the fact, that a plan, however carefully made, shows only *horizontal clearances* at the level where the plan section is taken, and hence a serious conflict above or below this plane may readily occur unless a *vertical section* is considered simultaneously. It is not always necessary to draw this vertical section, but it must be visualized and considered constantly in laying out the equipment. In all cases of doubtful interpretation of a single plan view it is advisable to show also one or more elevations, or, better still, an *isometric sketch* of such apparatus or connections.

Standard Symbols. The use of representative symbols to show standard features of the equipment is very common in all piping and wiring drawings and Figs. 1, 2 and 3 show the more common piping conventions used by the *U. S. War and Treasury Depts.* It is no longer customary to show piping or fittings by double lines in the general plan layout—time and space do not permit. For standard wiring symbols see chapter on "Electrical Equipment," Section 5.

The authors believe it to be entirely possible to show all piping lines, valves and fittings by the simple symbols shown herewith. A few applications of these symbols to heating practice are shown in Figs. 1, 2 and 3, and to ventilating practice in Figs. 4, 5 and 6.

Each engineering office will develop its own symbols as occasion requires, but it is advisable to keep such a *symbol system as simple as possible,* and remember that an occasional note is to be preferred to some rarely used and specially created symbol, the meaning of which may afterward prove a mystery to the reader.

All plans should carry, for purposes of identification, a *title, sheet number, date,* and the *initials,* at least, of the designer, the checker, and the draftsman. It is a great convenience to indicate on each sheet the *number of sheets* in the set, and all alterations or revisions of the completed drawings should be indicated by marking the *date of such revision* on the sheet.

SPECIFICATIONS, GUARANTEES AND PROPOSALS

After the plans are completed, or at least well started, and the type and capacities of the principal parts of the equipment have been decided upon, the preparation of the specifications should be begun.

These *specifications and the plans* are supposed to be *supplementary* to each other and the particular plans referred to should be *fully identified* in the specifications.

The specifications usually begin with a series of general paragraphs, often referred to as the *General Conditions,* which deal with (1) the kind and quality of materials, (2) samples, (3) approval of material, (4) modifications in the work, (5) inspections and tests, (6) patents, (7) permits, (8) guarantees, and (9) such features of the work and the bids or proposals to perform same as are generally applicable to any installation. A typical set of general conditions taken from the practice of the *Supervising Architects' Office of the U. S. Treasury Dept.* is given herewith, and illustrates by its specific requirements and restrictions the method by which that office endeavors to provide for general contingencies and secure uniform bids, with a reasonable freedom in the selection or approval of satisfactory materials and equipment.

General Conditions. (This Section of the Specification Includes the Plumbing, Gas Piping, Heating Apparatus, and Conduit and Wiring System.)

Kind and Quality of Material. All material, appliances, and fixtures furnished must be in strict accordance with the specification requirements in each case, and of the best quality and grade.

Bidders must furnish on the proposal sheet the information required thereby as to name and address of manufacturers and catalog number of trade name of material, appliances, and fixtures they propose to supply, and, where required by proposal sheet, give the names of *three different manufacturers* of the material, appliance, or fixture. The department reserves the right to select any one of the makes named, which selection is without prejudice to the goods of other manufacturers named and must not be otherwise construed.

It is the intention of this office to install a line of plumbing fixtures which are the *complete* product of one manufacturer, and in the consideration of proposals preference will be given (other things being equal) to the proposal which proffers fixtures in accordance with the intention above expressed. [The term "complete" as used in this paragraph is to be understood as applying not only to the fixture proper, but as including all valves, traps, fittings, supports, etc., required in its setting and equipment.]

In the event the successful bidder fails to comply with any of the requirements of the proposal sheet or of the preceding three paragraphs relative to material, appliances, and fixtures—*i. e.,*

(1) Fails to fill out the proposal sheet with names of manufacturers, etc., of appliances, etc., which he desires to use in the work;

(2) Fails to give the names of *three different manufacturers* where required on proposal sheet, even if the one or two named comply with the specification;

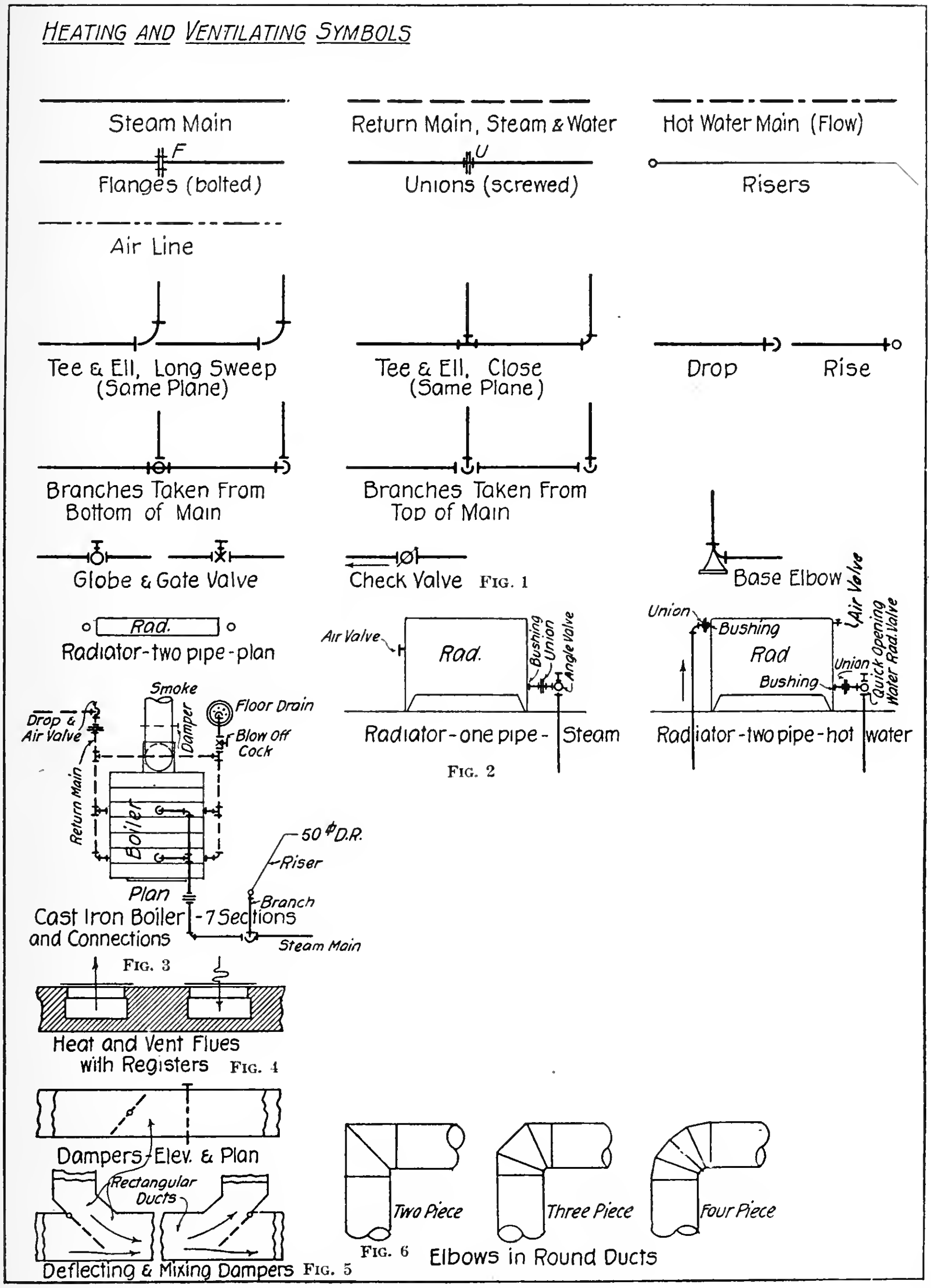

HEATING AND VENTILATING SYMBOLS
Steam Main
Return Main, Steam & Water
Hot Water Main (Flow)
F
Flanges (bolted)
U
Unions (screwed)
Risers
Air Line
Tee & Ell, Long Sweep (Same Plane)
Tee & Ell, Close (Same Plane)
Drop
Rise
Branches Taken From Bottom of Main
Branches Taken From Top of Main
Globe & Gate Valve
Check Valve
Fig. 1
Base Elbow
Rad.
Radiator-two pipe-plan
Air Valve
Rad.
Bushing
Union
Angle Valve
Radiator-one pipe-Steam
Fig. 2
Union
Bushing
Air Valve
Rad
Bushing
Union
Quick Opening Water Rad. Valve
Radiator-two pipe-hot water
Smoke
Damper
Floor Drain
Drop & Air Valve
Blow Off Cock
Return Main
Boiler
50 ϕ D.R.
Riser
Plan
Branch
Cast Iron Boiler-7 Sections and Connections
Steam Main
Fig. 3
Heat and Vent Flues with Registers
Fig. 4
Dampers-Elev. & Plan
Rectangular Ducts
Deflecting & Mixing Dampers
Fig. 5
Two Piece
Three Piece
Four Piece
Fig. 6
Elbows in Round Ducts

Fig. 1 to Fig. 6.

(3) Names appliances, materials, or fixtures not strictly in accordance with specification requirements in regard thereto or which are not of best quality and grade; or

(4) Fails to name a full line of plumbing fixtures which are the *complete* product of one manufacturer—the department reserves the right to reject any or all the material, appliances, and fixtures named by said bidder and to select those to be used in the work; which selection shall be final and binding upon the contractor, who must install for the contract price the material, appliances, and fixtures so selected.

The contractors must supply the superintendent of construction at the building with such cuts, catalogs, and descriptive matter as will enable him to identify readily the material furnished, and such cuts, catalogs, and descriptive matter, etc., furnished him must correspond in all particulars with material, etc., approved by the department.

Samples. The contractor must furnish for the approval of the Supervising Architect all the samples hereinafter called for, and also, if required by the Supervising Architect, samples of any or all of the material, appliances, and fixtures he proposes to use, and must pay all shipping charges on samples. No material, appliances, or fixtures of which samples are required to be submitted for approval will be permitted to be placed in the building until such approval has been given by the Supervising Architect.

Samples must be accompanied by a letter of transmittal from the contractor, and each sample must be marked with the name of the contractor and the name of the building to which it relates, and if of an article of an approved manufacture to have manufacturer's name also. Approved samples, if properly tagged so that same may be identified at time of final inspection, may be used in the work after serving their purpose as samples. Rejected samples of value will be returned to the contractor by express at contractor's expense, unless the letter of transmittal expressly states what disposition should be made of rejected samples.

Especial attention is called to the fact that no samples or drawings of any description in connection with the mechanical equipment are to be submitted unless specifically called for in the specification or by an office letter.

Approval of Material, etc. The approval by the department of any material, appliance, or fixture named on the proposal sheet is general only and is to be understood as an approval of same only upon its complete conformity with the specification requirements in regard thereto, and not as an absolute acceptance of the article.

Especial attention is called to the fact that the approval of a fixture by catalog number must be construed as applying only to fixture proper, all trimmings, fittings, etc., of fixtures to be in strict accordance with specification requirements.

Modifications, etc. It must be distinctly understood that the right is reserved by the department to require the contractor to make any omissions from, additions to, or changes in, the work or materials herein provided for when, in the judgment of the Supervising Architect, such are desirable. In the event the contractor fails to submit a satisfactory proposal for such omissions, additions, or changes, the Supervising Architect shall fix the price and his decision shall be binding on both parties, and no claim for damages on account of such omissions, additions, or changes, or for anticipated profits, shall be made or allowed.

Inspections and Tests. Upon completion of the plumbing, gas piping, and electric conduit and wiring systems, and on completion of heating apparatus except application of non-conducting coverings, the contractor shall give written notice to the Supervising Architect, through the superintendent, of his readiness for inspection and tests. Tests to be made in the presence and under the direction of the department's authorized representative. All labor, fuel, appliances, etc., necessary to make tests (except instruments for wiring test and expenses of department's representative) must be furnished and paid for by the contractor.

Should the inspection or test not be begun, through no fault of the contractor, within ten (10) days of the receipt of notice by the Supervising Architect, allowance will be made as hereinbefore provided.

Should the inspection or test be delayed upon the arrival of the inspector, or require repetition for any reason for which the contractor is responsible, the cost of delayed or subsequent inspections and tests, including the expressage on instruments, salary, and the traveling and other expenses of the inspector, shall be at the expense of the contractor and be deducted from any money due him upon the contract.

In all questions as to the interpretation of drawings and specifications, the satisfactory completion

of the work, and the defects necessary to be remedied, the decision of the Supervising Architect shall be final and binding upon the contractor.

In the event the contractor does not within a reasonable time remedy all defects and make all changes demanded by the Supervising Architect to complete the work satisfactorily, the right is reserved to have defects remedied or changes made, and to charge the cost of same against the account of the contractor.

No interpretation of drawings and specifications will be made prior to the award of contract.

Permits, etc. The contractor must obtain all permits, pay all fees and charges for connections to sewers, water mains, gas mains, etc., and use of public or private property for storage of materials, etc., and comply with the regulations of the city and local companies controlling these services relative to excavations, connections, repairing of surfaces, etc., outside the boundaries of the government property. The local regulations in regard to plumbing, etc., do not apply to work inside of the lot line.

Guarantee. The bond which will be required in connection with the contract must guarantee each and every part of the mechanical equipment. The contractor will be required to remedy at his own expense all defects which may develop by reason of the use of any inferior or defective materials or workmanship. It must also be understood and agreed that the final acceptance and payment for the work will not relieve the contractor for having installed defective materials and work not apparent at time of final inspection.

Patents. The department will not recognize demands brought on account of infringement of patents, but will hold the contractor and his bondsmen strictly responsible for any delay or any cost resulting from his failure to protect the Government fully against patent rights.

Specifications. The specifications proper must include a brief *description of the proposed installation,* confined to a few paragraphs. This preliminary statement is of considerable importance to the bidder, as it should give him in concise terms a general idea of what is intended by and shown in detail on the plans, and more completely described in the specifications.

Each part of the equipment to be furnished and installed is then specified in detail in the *body of the specifications,* and so identified as regards size and capacity that no confusion may arise as to just what is intended. The greatest care must be exercised to give all *limiting conditions* such as pressures, speeds, temperatures, fuel values, weights, etc., under which each part of the equipment must operate. A typical specification will be found in the Appendix.

Guarantees. The question of *guarantees,* if any are required, must be taken up at this time, and the items, such as coal, oil, or gas consumption, water evaporated, steam used, water or air required, and efficiencies all relating to the economical or satisfactory performance of the equipment, specified in such a way that the bidder may submit the necessary *guaranteed performance data* to cover the range desired.

These guarantees usually require that an *acceptance test* of such equipment be made, and definite *penalties* are usually specified in case of failure to meet or equal the guarantee submitted. Such penalties may amount to *entire rejection* of the deficient apparatus, or a *percentage deduction* may be made from the contract price in proportion to the extra cost of retaining and operating such equipment.

Proposals and Contracts. Advertised notice of the construction of a new building and its equipment is usually given in the engineering journals or otherwise, and the bidders on the work are required to submit *formal proposals* protected by a *guarantee* or certified check for 10 per cent of the amount of their bid.

This guarantee is intended to show good faith and exclude irresponsible parties from the competition, as it is generally assumed the *contract* will be *awarded* to the lowest responsible bidder, and that he will then agree to sign a contract and give a bond for its fulfillment.

Each proposal must be *absolutely regular,* that is, must furnish all data, such as performance guarantees, names of manufacturers and trade name, and designation number of apparatus the bidder proposes to furnish. This last requirement is especially important in all cases where the specifications permit a choice between several makes of equipment, or say "*equal to.*" Unless the bidder intends to use the apparatus actually named in the specification he *must name*

on his proposal sheet just what type, make and size of equipment he will use in place of it. In no other way can awards be made with any degree of justice, or assurance that satisfactory apparatus will be subsequently installed.

Proposals that are not regular or incomplete should be *rejected* at once.

Unit prices are often called for in the proposal to cover such cases as may arise when the exact amount of excavation, pipe and duct work, etc., cannot be definitely determined in advance, but where the general character of the work is known. In these cases the cost per cu. yd. or per lineal ft. or per pound is stated and an *extra* or a *credit* allowed, depending on whether more or less work than shown on plans is required.

Alternate proposals, or bids on alternate requirements, are very often called for. In this case it may be necessary to provide in advance for cutting out or cheapening certain features of the installation if the available funds are limited. This contingency—the acceptable bid exceeding or even falling much below the funds available for the work—must be anticipated, and by calling for suitable alternate proposals it is often possible to save readvertising the work with consequent loss of time, and at the same time make a reasonably satisfactory award. Every contract of any consequence should be protected and its fulfillment guaranteed by the filing of a *contractors' bond* as soon as the award is made; in fact, the award is usually made effective only when such bond is filed. The bond usually amounts to 50 per cent of the contract price.

COST OF EQUIPMENT IN GENERAL

General Conditions. The cost of the mechanical equipment for buildings varies with the general commercial and industrial conditions throughout the country, and also varies with the locality in which a given building is placed. Localities remote from production and labor centers must pay more than others more favorably situated, and hence it is quite impossible to give a schedule of prices or costs which is universally applicable for any considerable period of time.

Nevertheless *estimates* must be made on work for which the plans and specifications have been prepared, and the contractor must have constantly available a mass of cost data both of a *general* and *detailed* nature in order to make an intelligent bid.

The engineer must, of course, make his own estimate in order to arrive at some basis for comparing the various bids submitted, and two methods are employed, depending for their selection upon the time to be devoted to making the estimate, or the accuracy desired.

General estimates are of course most readily made and are most logically based on the cu. ft. of space in the building or upon average unit prices per fixture or per unit of capacity in the apparatus, such as horsepower or kilowatt, or in many cases on the weight in pounds of certain equipment, such as sheet metal ducts.

Detailed estimates require that all quantities be carefully measured and properly tabulated as to kind, size and number, and then the current *net prices* plus *freight* must be determined for each kind of equipment concerned, and finally a suitable *labor charge* must be allowed for making the actual installation, and a reasonable *profit* of from 15 to 20 per cent added to the total cost so determined. The *net* price is determined from the manufacturer's published *list* price, as given in his price lists or catalogs, by ascertaining the current *discount* allowed for that particular article and modifying the list price accordingly.

Estimating Data for Mechanical Equipment. A large amount of both general and detailed estimating or cost data relating to mechanical equipment is given by *Nelson S. Thompson* in his valuable book on the "Mechanical Equipment of Federal Buildings." These data, of course, are based on government work and will apply only to the very highest grade of commercial installations, and therefore will prove amply safe when applied to work of average grade.

COST OF HEATING AND POWER-PLANT APPARATUS

Cost data, more generally applicable to the average commercial installation, are given in the following paragraphs from an article by *W. J. Downing* in "Power," November 13, 1913. These data are based on *cost to the contractor* and do not include his profit:

"In order that engineers and owners may form an opinion as to the probable cost of a proposed system, the following information is submitted: List prices of material are usually available, and when possible discounts are given for standard apparatus. All prices are based on actual installations, most of them in the New England States, and allowance should be made for other localities, based on the difference in cost of labor and material.

"**Radiation.** Radiation will be classified under five headings:

"1. Cast-iron direct radiators cost 19c. to 27c. per square foot of surface, depending on the height of the radiator. The labor cost will be nearly the same for casting and finishing a section containing 1 sq. ft. of surface as for a section containing 5 sq. ft.

"2. Cast-iron indirect radiators of the pin type for gravity work cost 16c. to 18c. per square foot.

"3. Cast-iron radiators for fan systems cost 25c. per square foot.

"4. Pipe coils for direct radiation cost 30c. per square foot.

"5. Pipe heaters consisting of 1-in. pipes with cast-iron bases for fan systems cost 45c. to 50c. per square foot of surface. For cast-iron bases with a damper for direct-indirect radiators add $1.25 for each 10-in. length of base.

"The labor cost for installing direct radiators on a one-pipe system can be obtained by allowing one day's time for a steam-fitter and his helper for each radiator. This covers the time required to run the vertical risers and connect and set the radiators. It does not include the time required to place the horizontal mains in the basement and connect up the boilers. This item will be covered under another heading. For a two-pipe system allow 1½ days' time for a fitter and helper per radiator.

"Indirect radiators for gravity and fan-blast systems cost about ½c. per pound for the former and 1c. per pound for the latter for erection, together with the labor cost of a fitter and helper for one day for each four connections made to the heater sections.

"Allow 2½c. to 3c. per square foot of surface of pipes and radiators for bronzing.

"Automatic air valves cost 75c. to $1 each in place.

"For temporary setting of direct radiators used to furnish heat in the building, while under construction, allow for each radiator.

"Figures based on a large number of installations show that an allowance of $50 per thermostat should be made for automatic control. This includes the air piping, compressor dampers and thermostats, set in place and connected.

"**Boilers and Auxiliaries.** Small cast-iron fire-pot boilers for house heating cost $30 to $35 per square foot of grate area.

"Cast-iron sectional boilers for house and public-building heating cost $21 to $25 per square foot of grate area.

"Horizontal fire-tube boilers set in place complete with trimmings ready for steam and water connections cost $12 per horsepower.

"The Manning type of vertical boiler for power-plant work will cost $10 per horsepower erected.

"Water-tube boilers set in place with trimmings cost $14 to $16 per horsepower.

"Internally fired boilers of the Morrison type cost $16 to $18 per horsepower, including trimmings.

"Dutch or extended ovens are often used in power plants for burning a low grade of fuel, or utilizing the waste material from manufactured products. These ovens will cost $250 for a 300-horsepower unit.

"Superheaters cost $2.25 to $3 per horsepower, depending on the size and type.

"Special boiler settings designed to economize heat, similar to the Smith setting, cost about $150 per boiler.

"All of the above prices are based on boilers with plain grates. Shaking grates should be figured at from $5 to $6 per square foot of surface.

"Feed-water heaters of the closed type cost from 75c. to $1 per horsepower, depending on the size of the unit. Feed-water heaters and purifiers of the open type cost $2.20 per horsepower for a 100-horsepower unit and $1 per horsepower for a 1000-horsepower unit. Intermediate sizes cost a proportional amount.

"A good damper regulator for controlling the draft in boilers can be obtained for $50.

"Boiler-feed pumps cost 50c. per horsepower capacity of units of 150 to 200 horsepower.

"Blowoff and return tanks suitable for 100 lb. pressure cost about 8c. per pound in weight.

"Copper hot-water tanks good for 100 lb. pressure complete with steam coil cost about $1 per gallon capacity. Add $50 if the tank has automatic control.

"Steam traps take a discount of 40 per cent from list prices.

"**Pipe, Fittings and Valves.** While there are several large manufacturers of these products it is usually safe to figure the following discounts: Steam pipe, 75 per cent; valves, 50 to 60 per cent; cast-iron fittings, 70 per cent; spiral-riveted pipe, 70 per cent.

"An accurate list should be made of the actual material required for any particular installation, as there are too many variables to use a unit price per horsepower capacity of the plant. The labor cost will average $1.50 per horsepower for connecting the boilers and installing the basement mains in plants of 200 to 400 horsepower.

"The special valves necessary for a first-class vacuum system cost $6 to $8 per radiator. Another method of figuring vacuum systems is to allow 10c. per square foot of radiation for the special apparatus required.

"**Covering.** An asbestos covering 4 in. thick for boilers and heaters will cost in place 50c. to 60c. per square foot of surface. Air-cell covering 1 in. thick will cost 22c. per square foot. Eighty-five per cent magnesia 1 in. thick will cost 30c. per square foot. These prices include the labor required to apply and are useful in calculating the cost of covering heating ducts and smoke flues.

"Steam-pipe covering made of 85 per cent magnesia will cost one-half of the list price, including the labor of applying. If desired the discounts applying to the various types of covering can be obtained and the labor cost based on the fact that one man will cover 100 ft. of straight pipe per day up to 4-in. diameter or will cover 40 fittings per day up to 4-in. size. The above amounts will be more for larger sizes due to the increased labor of handling.

"**Ventilating Apparatus.** Centrifugal steel-plate fans for ordinary systems in which the total pressure does not exceed ¾ oz. will cost $10 to $13 per 1000 cu. ft. of air per min. capacity, depending on the size.

"Direct-current motors for driving fans will cost $18 to $25 per horsepower. Regulating rheostats cost 60 per cent. of the list prices.

"High-pressure engines for fan driving cost $10 to $16 per horsepower. Low-pressure engines for fan driving cost $18 to $22 per horsepower.

"Air washers are usually based on a velocity of 500 ft. per min. and on that basis cost $18 to $26 per 1000 cu. ft. of air per min. capacity. Erection of fans, motors and air washers will cost about 1c. per pound in weight.

"**Galvanized-Iron and Steel-Plate Work.** Piping arrangements employing galvanized-iron distributing ducts cost about 15c. per pound in place. The ratio of weight of iron to the cubic contents of the building varies widely with different types of building. In factory work where heating is the primary object the galvanized-iron ducts for an overhead system will average 1 lb. of iron to 100 to 125 cu. ft. of contents. In buildings where ventilation is the main object no standard values can be given, as the amount of metal will depend on the standard of ventilation maintained. In each case the actual weight of metal must be calculated from the plans.

"Steel-plate work for smoke flues costs from 6c. to 8c. per pound.

"**Registers and Screens.** Cast-iron registers for floors and side walls cost one-fourth the list price. Bronze registers cost one-half the list price. Plain wire screens with angle or channel-iron borders cost 15c. to 25c. per square foot. Allow 3c. per square foot for bronzing.

"Filter screens of cheese cloth for removing dust from the air are based on a velocity of 30 to 50 ft. per min. through the net area. Their cost will be from 50c. to 70c. per square foot, depending on the quality of material. Mushroom ventilators cost 65c. to 75c. each.

"**Foundations.** Allow 75c. per cubic yard for excavation in ordinary soil and $4 per cubic yard for rock. Brick foundation walls cost 40c. to 50c. per cubic foot in place. Concrete foundations cost $6 to $7 per cubic yard for the concrete and 15c. per square foot of surface for the forms. Water-proofing will cost 40c. per square foot.

"**Sprinkler Systems.** Sprinkler systems cost from $3 to $3.25 per head, including pipe, sprinkler heads and erection. Hose racks for fire protection in public buildings cost $50 each, including piping and erection.

"**Gas Piping.** In fireproof buildings gas-pipe systems cost $5 to $6 per outlet for labor and material. For residences of the usual frame construction allow $2.50 to $3 per outlet.

"**Electric Lighting.** Labor and material for wiring will cost about $6 per outlet. Three-light office fixtures will cost $7 each, two-light brackets $6, and one-light brackets $4. These prices are for a good, substantial fixture hung and connected.

"**Unit Costs.** While the conditions of various installations make it impossible to give a unit price for a system that will apply in all cases, the average of a large number of jobs shows some interesting results. The average cost of a heating system for dwelling houses using direct-steam radiation is 80c. per square foot of radiation. For office and factory work allow $1 per square foot of radiation. For hot-water direct radiation allow $1.25 per square foot of radiation. To these prices should be added that of the boilers to obtain the cost of the entire system.

"Although the size of direct-steam radiators varies over a wide range the cost of complete systems, exclusive of boilers, averages $37 per radiator.

"All prices stated in this article are the costs to the contractor. An allowance for contractors' profit should be added to the total cost of the system. Profit is usually figured as a percentage of the total cost and will vary from 10 to 15 per cent. It will be noticed that the prices stated above give a considerable range and the question may arise as to the exact value to be used. It may be helpful to note that in any case a price should be selected depending on the size of the apparatus. For instance, a boiler with 5 sq. ft. of grate area will cost more per square foot than one with 20 sq. ft. By paying attention to the relative size of the unit in question a fair estimate can be made of the cost from the values given."

COST OF HEATING AND VENTILATING SYSTEMS FOR SCHOOLS

Unit cost data, based upon *cubic contents*, are of great value if carefully applied to buildings of the same type as those for which the data were originally determined. *D. D. Kimball* has reported such data in the "School Board Journal," and the following paragraphs, taken from this article, give the unit volume costs for heating and ventilating systems in five classes of schools:

"In a study of the cost of installation of heating and ventilating plants, made in a number of schools, it was found that the prevailing custom of apportioning a certain percentage of the total cost of the building for the installation of the heating and ventilating plant is of no value, as these percentage ratios vary more than 100 per cent, even with similar classes of installations. For a given size of building, the cost of the heating and ventilating system will be approximately the same whether the building is a monumental stone structure or a plain wooden structure, but the percentage of cost of the system will be very different.

"**Classification of Systems.** As a result of this study, the following scheme of classification has been arrived at:

"Class A. Plants providing for fire-tube boilers, double fan systems, air washers and humidi-

fiers, individual or double duct systems and modulating control of direct radiators and mixing dampers.

"Class B. Same as Class A, but using automatic stokers and water-tube boilers instead of fire-tube boilers.

"Class C. Same as Class A, but eliminating the modulation control of radiators and dampers and using the single trunk ducts.

"Class D. Same as Class C, except that it eliminates the use of air washers and humidification systems.

"Class E. All other systems.

"Manifestly there are many combinations of equipment which render an exact determination of classification difficult, but in general this classification has proven satisfactory.

"After a careful study of this method of classification and the figures on costs as thus obtained, it was found that the only satisfactory basis of determining the cost of the installation of the heating and ventilating plant was on the basis of the cubic feet of space in the building. The variation in costs within the different classes of systems is rarely over 10 per cent from the average, the greatest variation occurring in Class A. The resulting costs are as follows:

"Class A, cost of plant per cu. ft. 2.7c. to 3.3c.—average 3.1c.

"Class B, cost of plant per cu. ft. 3.3c. to 3.8c.—average 3.4c.

"Class C, cost of plant per cu. ft. 2.2c. to 2.5c.—average 2.4c.

"Class D, cost of plant per cu. ft. 2.2c. to 2.3c.—average 2¼c.

"Class E, cost of plant per cu. ft. 1.9c. to 2.2c.—average 2.1c.

"If classes D and E were but abandoned and a proper amount of skill were used in the design, installation and operation of the remaining classes, a sufficient appropriation being provided for the installation and operation of the ventilating plant, it is believed that little basis would be left for complaint as to the success of the artificial ventilating system.

"Classes D and E are the result of a too limited appropriation, a demand for too large a building for the funds available, too much ornamentation, or too much equipment, or, in other words, an attempt to build a $100,000 building with a $75,000 appropriation, the greatest sacrifice being made in connection with the heating and ventilating plant. Better were a proper building, well equipped, though smaller.

"As a matter of information it is interesting to note that the cost of plumbing equipment for school buildings ranges from three-quarters of a cent to one and one-half cents per cubic foot, the average being one and one-tenth cents. The cost of electrical equipment, exclusive of electric power plants, ranges from one-half to one cent per cubic foot, the average being seven-tenths per cubic foot.

"In the case of the heating and ventilating, plumbing and electrical work, the costs seem to be approximately the same in grade schools and high schools."

APPENDIX I

The Gale Centrifugal Dust-Collecting System.* A typical exhaust fan system for handling shavings, sawdust, etc. is shown by Fig. 1. The elements of such a system are: (1) the hoods attached to the machines to catch the refuse as it is thrown from the knives or saws; (2) the exhaust fan for creating the suction and conveying the material; (3) and the cyclone separator, or dust collector, into which the air and dust are discharged by the fan. The function of the separator is to remove the solid material from the air by centrifugal action.

The Gale Centrifugal Dust Collector and Air Washer was recently developed by the *Oneida Steel Pulley Co.*, Oneida, N. Y., to replace the ordinary cyclone separator. This washer (Table 3) is connected directly to the fan, thus saving discharge pipe, reducing friction losses, and permitting the recirculation of the air from the exhaust system. A plan view of this washer is shown by Fig. 2. The dust-laden air from the fan enters at *A* with a high velocity, 3500 to 4000 ft. per minute. As it sweeps around the curve of the washing chamber *B*, the refuse and dust are thrown against the wet, curved side of the machine, and are washed to the bottom by the spray water. The clean air then whirls around the body of the washer, *C*, and out of the opening *D*, after eliminating all free water by the same centrifugal force that separated the dust from the air. The muddy water and refuse flow around the bottom of the body to the outlet *E* and thence to a strainer or to the sewer.

The final disposition of the refuse is, of course, different with different materials. Shavings are usually used as fuel, being fed to the fires through automatic furnace-feeders, as in Fig. 1. Shavings and sawdust are also baled or bagged, to be sold. Polishing and grinding dust is frequently valuable for the emery or the metal it contains, especially in the jewelry trades.

To convey various materials through sheet-metal pipes widely varying velocities of air are required. Light, dry refuse, in small pieces, can be kept in motion with low air speeds, while damp, sticky materials, or large, heavy chunks frequently collect in the ducts, even at high speeds. Table 1 gives the velocities necessary to handle the materials most frequently encountered in "blow-pipe" work.

TABLE 1

VELOCITIES REQUIRED FOR CONVEYING MATERIAL

Material	Air Velocity Ft. per Minute
Pulverized colors (paint)	1500
Flour	2000
Wood sander dust	2000
Sawdust, dry	3000
Sawdust, damp, or green	3500–4000
Shavings, dry	3000
Shavings, damp, or green	3500–4200
Polishing dust	3000
Buffing dust, dry	3000–3300
Buffing dust, sticky, rouge	3500–4000
Hemp, Jute, Sisal	4500–5000

* The authors are indebted to *J. L. Alden* for this description and the cuts illustrating the operation of the system.

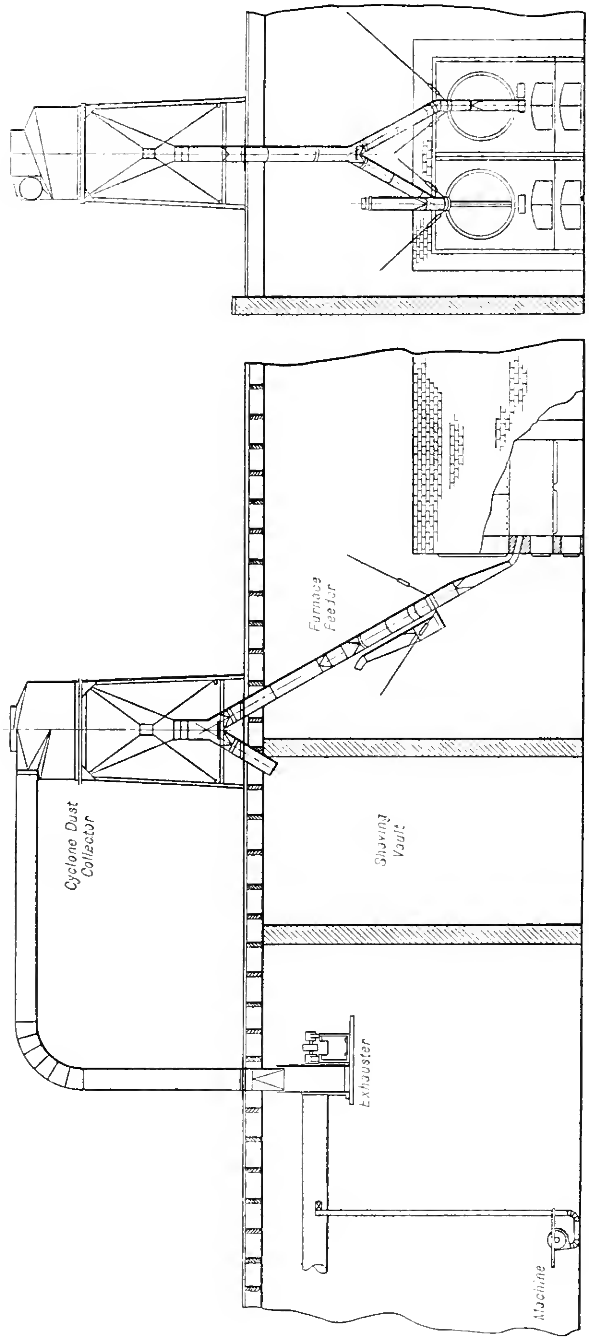


FIG. 1. GALE CENTRIFUGAL DUST-COLLECTING SYSTEM.

The power required in exhaust work is much greater than that usually encountered in ventilating systems. A static suction of from 1 inch to 4 inches of water must be maintained at the hoods; high velocities are usually necessary; the exhausters used are, as a rule, of an inefficient type, and the resistance of the ordinary centrifugal separator is high, from 1 inch to

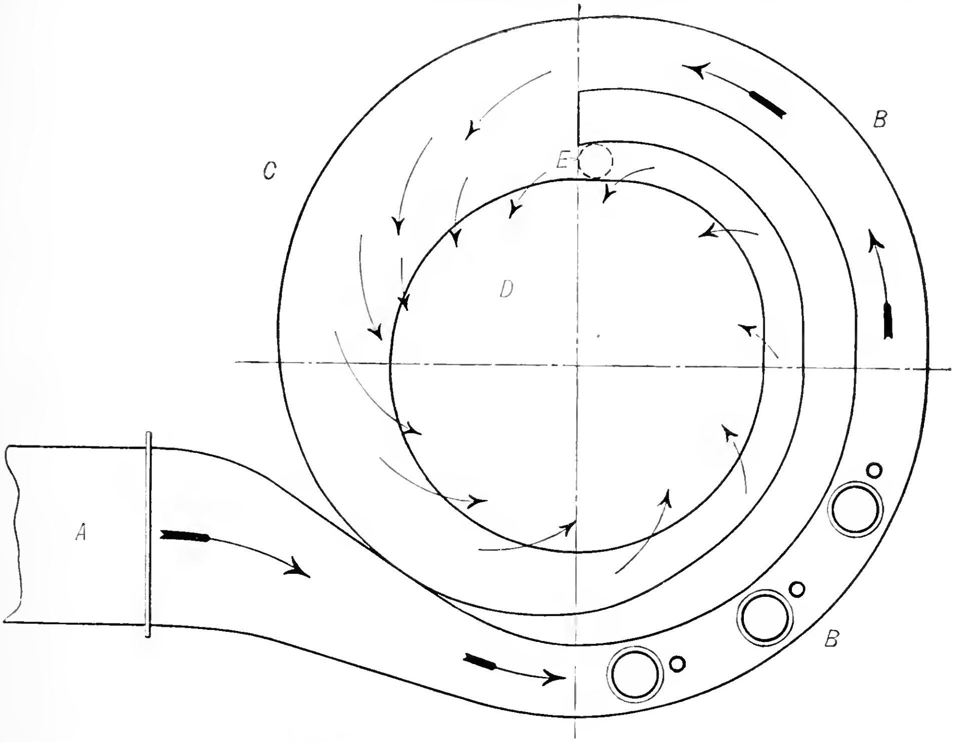


FIG. 2. PLAN VIEW OF GALE DUST COLLECTOR AND AIR WASHER.

7 inches of water. Table 2 gives efficiencies of the ordinary types of exhausters used in this class of work. The higher figures cover the most efficient conditions, when the fan is connected to branch pipes having a total area of approximately the "blast area" or "capacity area" of the fan. The lower figures are those for a fan piped to the full inlet area, or about twice blast area.

TABLE 2

AVERAGE EFFICIENCIES OF EXHAUST FANS

Type	Efficiency Per Cent
Radial blade, 6-blade	22–45
Radial blade, 8-blade	27–45
Curved blade	35–50
Multi-blade (suitable only for pulverized material)	25–60
Tubular wheel	60–75

NOTE.—Ball bearings save from 8 per cent to 12 per cent when handling air and light materials, and from 15 per cent to 22 per cent when used on conveying fans handling jute, hemp, and similar heavy materials.

In general, when buying dust-collecting equipment, a binding guarantee with a forfeit attached should be demanded. This guarantee should cover the vacuum at the fan, the power input to the fan, and the resistance or "back pressure" imposed by the separator or receiving device.

TABLE 3

DIMENSIONS OF GALE CENTRIFUGAL AIR WASHERS

Capacity Cu. Ft. per Min.	Greatest Diameter Inches	Height Inches	Weight Pounds
1000	48	18	125
3000	56	24	150
5000	68	32	175
7000	72	36	200
9000	80	40	250
11000	86	44	350

NOTE.—The water consumption is approximately 2 gallons per 1000 cu. ft. of air per minute. This water can ordinarily be recirculated.

APPENDIX II

SPECIFICATIONS FOR STEAM HEATING A FACTORY OFFICE BUILDING

Refer to Pages 241 and 597

(See General Conditions, Page 594, which will govern for Steam Heating where applicable.)

Description. 1. The work covered by these specifications and the plans consists in furnishing all labor and material necessary to install a steam-heating apparatus complete in the factory office building.

2. The apparatus is to be arranged for a one-pipe low-pressure circulating system; the steam being supplied by the boiler located in basement, and distributed to the radiators through a system of piping, as shown on separate heating plan.

3. Apparatus when completed must be entirely free from all hammering and cracking noises when in operation.

4. All work and material must be properly protected, and before the apparatus is accepted the contractor is to put every portion of the work in first-class condition, at his expense, and will make good all damage to the building caused by his workmen, and remove all rubbish resulting from his operations from the premises.

Boiler, Boiler Trimmings, Etc. 5. *Boiler.*—Contractor shall furnish and set up complete in all respects in the basement, where shown, one horizontal return-flue cast-iron sectional steam boiler, equal to the Ideal S-25-7 of not less capacity than 1600 square feet of direct radiation. Contractor to provide suitable concrete foundation and line the ash pit or under grate surface with hard burned brick.

6. The boiler must be made of best quality tough gray cast iron, entirely free from sand holes or other defects, and must be provided with all necessary castings, including base, grate, soot doors, check draft for chimney, shaker, etc.

7. *Grates.*—The boiler must have a grate surface of not less than 8.16 square feet. The grates must be heavy, of approved rocking and dumping pattern, furnished with lever handle and with some device on the shaking apparatus outside of the boiler to indicate positively that the grates are in proper position after shaking.

8. *Name Plate.*—The boiler must have the manufacturer's name and address, together with its trade name and catalog number, distinctly cast on its front or on a standard name plate of the manufacturer, which shall be securely fastened on front of the boiler. The name of the distributing agent will not fulfill the above requirements.

9. *Instruction Card.*—The contractor must provide and place in a convenient position near the boiler a printed card showing manufacturer's instructions for cleaning and operating the same.

10. *Boiler Fixtures.*—Furnish and attach to boiler all necessary fixtures, consisting of steam and water gage, gage cocks, water column, safety valve and damper regulator as hereinafter specified.

11. *Steam Gage.*—One 5-inch iron case with brass rim, improved full Bourdon, or equal, single spring steam gage, with non-corrosive movement, black dial and light figures. This gage must be provided with gage cocks and siphon.

12. *Water Gage.*—One $\frac{5}{8} \times 10$-inch water gage, with finished brass fittings, four guards, bottom waste cock, and top and bottom brass valves with hardwood handles.

13. *Gage Cocks.*—Three gage cocks with soft metal seats, stuffing boxes and hardwood handles.

14. *Water Column.*—Ornamental cast-iron water column, tapped for the above-mentioned water gage and gage cocks, and connected to the boiler by means of $\frac{3}{4}$-inch heavy brass pipe, with brass tees, ells and plugs so arranged that it can be cleaned by means of a stiff rod. If boiler has column cast on front section, same will be accepted.

15. *Safety Valves.*—One 2-inch all brass, improved pop safety valve, with side outlet, set for

15 pounds pressure after the test pressure of 20 pounds has been made as hereinafter specified. This valve is to be connected in manner shown on plan.

16. *Automatic Damper Regulator.*—One improved low-pressure, sensitive, positive-acting damper regulator, constructed entirely of metal, to be attached to the boiler above the water line. The pipe connection to be provided with a brass gate valve. The regulator to have brass chains and pulleys attached to the draft door and damper or check draft in smoke pipe, and to be so adjusted as to regulate the draft and maintain the required pressure without attention.

17. *Tools.*—One full set of stoking and firing tools, consisting of one poker, one hoe, one slice bar, one suitable steel brush with jointed handle, one scoop and one straight-handled shovel.

18. *Water Connections.*—Two ¾-inch brass gate valves and one swing check valve with galvanized pipe and fittings to be connected by the heating contractor with supply pipe and main return line near the boiler. The check valve to be placed between the gate valves and in a horizontal run of pipe.

19. *Blow-off.*—Provide and connect to the lowest point of main return, where shown, a 1¼-inch blow-off pipe with brass cock and elbow as shown. The elbow to be above floor trap so that any leak will be visible.

20. *Boiler Test.*—Boiler must be tested to 80 pounds cold hydraulic pressure before leaving the shop. Certificate of inspection must be furnished, this certificate to be turned over to the owner at completion of the work. Boiler must be tested, when assembled in the basement, to 50 pounds hydrostatic pressure to ascertain the presence of cracks or other defects.

21. *Smoke Pipe.*—Connect the boiler to the chimney by means of a smoke pipe made of No. 16 U. S. S. gage black iron of dimensions shown, and place in pipe a shut-off damper with lever handle and adjusting attachment. Smoke pipe must be provided with ample sized clean-out at elbow, so that entire length of smoke pipe may be cleaned without taking down. Smoke pipe shall not be closer than 12 inches to ceiling of basement.

Connecting Pipes. 22. The above described boiler is to be connected to the radiators throughout the building by means of a one-pipe, circulating, gravity return system, as shown on plan. The supply main after leaving the boiler is to be run as close as practicable along ceiling of basement graded in the direction of the flow of steam, with a grade of approximately ¾ inch in 10 feet. All pipes shall be graded to outlets without forming traps. All branches to first and second floor radiators must be taken off separately from the top of the steam mains or at an angle of 45 degrees with elbows and nipples.

23. The low end of the longest steam main shall not be less than 18 inches above the water line of the boiler.

24. No returns or reliefs are to be connected together above water line of boiler.

25. All radiator connections and branches to have enough fall towards risers and mains to leave a good grade after pipes have expanded.

26. *Risers.*—All risers are to be run with offset fittings, if required, so that the pipes will be about two inches from the walls.

27. The location of risers shown on plan are approximate; the risers are to be so run that they will not cut the joists.

28. *Valves on Mains.*—Provide, where shown on plans, gate valves with double non-corrosive seats, same to be heavy weight and constructed for a working pressure of 125 pounds. Valves 2-inch diameter and smaller to be made of best quality steam metal with screwed ends, and those of 2½-inch diameter and over to have flanged unions, iron body with bronze mountings, including bronze stems and faces, and removable bronze seats. Valves 4-inch diameter and over shall have outside yokes and rising spindles designed to be packed under pressure. (*Note.*—This paragraph does not apply to single boiler installations where no valves are installed.)

29. *Expansion of Pipes.*—All pipe work must be so constructed that it will be free for contraction and expansion, so that it will not damage any other work or effect injury to itself.

30. *Pipe Supports.*—All steam and return pipes must be suspended with heavy adjustable

expansion pipe hangers of approved pattern, placed not over eight feet apart. No cold rolled flat steel hangers will be accepted.

31. *Flange and Combination Unions.*—On each long run of steam and return piping locate two ground joint combination iron and brass or flange unions, so that any part of the apparatus may be disconnected without injury to the remainder. Combination unions may be used on all pipes not exceeding two inches in diameter; all larger must be made with flange unions.

32. *Pipe Sleeves, Plates, Etc.*—Where pipes pass through wood floors and partitions, they are to be fitted with heavy galvanized iron adjustable thimbles, and in all finished rooms heavy nickel-plated cast-iron floor and ceiling plates must be provided. These plates must not be placed in position until just before the building is turned over to the owner. Ceiling plates must be securely held in place by set screws.

33. Where risers pass through cornices or curved surfaces, loose-fitting galvanized iron sleeves must be installed with the end cut to suit the profile and projecting $\frac{1}{4}$ inch from the finished surface, no ceiling plate being used.

34. Where pipes pass through brick walls in basement, wrought-iron pipe sleeves two sizes larger than the pipe itself must be installed, properly cemented in place.

35. Returns will be run in concrete trenches where they are carried below the floor line, as shown on plans. Pipes beneath the unexcavated portion of the building must not come within 12 inches of the ground on any side. Where necessary to accomplish this, trenches with a bottom width of 12 inches and side slopes of 1 to 1 must be dug.

36. *Pipe Covering.*—All pipes below the first floor must be covered with the best quality magnesia sectional covering not less than one inch thick, containing not less than 85 per cent of carbonate of magnesia, and all valves and fittings with plastic material of same thickness and composition as specified for pipe. The jacket of covering to be of 6-ounce canvas and the whole put on with No. 30 B. and S. gage solid sheet brass straps not less than 1 inch wide, and not more than 18 inches apart. It must be put on true and even in the best manner by skilled workmen, and all joints made tight with plastic magnesia. All coverings to be moulded in two sections and must have the manufacturer's label pasted on each length.

37. The contractor is requested to name in his proposal and submit a sample of heavy asbestos air-cell or asbestos felting covering, which he will install in lieu of the magnesia above specified, stating what amount is to be added to or deducted from his contract price should either of the latter be selected instead of magnesia. Samples to be submitted to the engineer for approval.

38. *Boiler Covering.*—Boiler, except fire and clean-out doors, must be covered with $1\frac{1}{2}$ inches of magnesia cement plaster containing not less than 85 per cent of carbonate of magnesia properly secured on 2-inch mesh galvanized iron netting and the finishing coat troweled to a hard smooth surface.

39. *Smoke Pipe Covering.*—Cover smoke pipe with three layers of 25 per cent asbestos paper weighing approximately $1\frac{1}{4}$ pounds to the square yard, layers to be wrapped on diagonally in opposite directions and secured with copper wire.

40. *Pipe.*—All pipe to be used to be new mild steel of best make, and of the National Tube Company's standard weights and sizes; all over $1\frac{1}{4}$ inches to be lap-welded. All pipe must have burr removed from the ends, and no bending of pipe will be allowed where fittings can be used.

41. *Fittings.*—All fittings used in this work are to be of size and weight to correspond with the above standard pipe, fine-grained gray cast iron, heavily beaded, with clear-cut taper threads, without a malleable fitting, lock nut, running thread or long screw in any part of the apparatus. Eccentric fittings shall be used where necessary to secure unobstructed flow of the water of condensation.

42. *Joints.*—All screw joints are to be made with long taper threads and made perfectly tight with the use of a stiff mixture of graphite and oil applied with a brush to the pipe only; in no case to the fitting. Flange joints to be made with the best quality of combination copper-ring and asbestos gaskets.

Heating Surfaces, Etc. 43. *Direct Radiation.*—The building is to be heated by direct radiation, same to consist of ornamental cast-iron two-column radiators, equal to the Rococo or Triton, of standard weights, 32 inches high, except where otherwise shown. Sections to be put together with heavy right and left cast-iron screw nipples in the best manner.

44. *Wall Radiators.*—Place, where shown, wall radiators consisting of the number of 7- or 9-foot sections indicated, all put together with right and left cast-iron screw nipples, and supported by suitable brackets or hangers as directed.

45. *Size of Radiators.*—Location of radiators is shown on construction plans, and size of radiators to be as shown on separate heating plan, and must in no case be less. Where bidder thinks more surface is required for exposed position, he will add amount needed to furnish the required amount of heat, and state in his proposal the cost of the extra.

46. *Valves on Radiators.*—Each radiator to be operated by one valve, with ball-joint union connection; valve shall be of best steam metal, extra heavy, with rough body, finished trimmings, heavily nickeled all over, and provided with polished hardwood handle. All radiator valves to be of the offset or corner type where required, and will be connected to radiators by ground brass nickel-plated unions so that any one radiator can be disconnected without interfering with the rest of the apparatus. Valves for wall radiators, where necessary, must be gate valves with unions of same make and finish as above. No globe valves to be used. All radiator valves to be equal to Jenkins Bros.' " Diamond Brand " or Fairbanks; radiator valves other than the above must be submitted to the engineer for approval. Radiator valves must be of same size as connection shown on plan.

47. *Air Valves.*—Each radiator must be provided with full nickel-plated improved automatic air valve equal to the Monash No. 6, with four-way drain, or the Norwall syphon, with the name stamped on the outside. Twelve keys to be furnished with valves requiring same. Ends of steam mains must be provided with a large size Breckenridge air valve, provided with shut-off cock between air valve and main.

48. *Painting.*—All radiators to be primed with yellow paint free from oil and finished in aluminum bronze. All pipes, castings, etc., before covering, to be painted with two heavy coatings of black graphite paint of the best quality and approved by the engineer. All pipes exposed in rooms are to be painted to match radiators and all pipe covering is to be painted three coats of lead and oil in lead tint.

49. *Cutting.*—Do all cutting except woodwork, and all boring necessary to install heating system. Any damage or defacement done to the building during the progress of the work is to be made good at the expense of the heating contractor.

50. *Workmanship.*—All work is to be done in a neat, substantial and workmanlike manner, and the apparatus is to be left in perfect working order, to the satisfaction of the engineer.

51. *Testing Apparatus.*—The contractor is to furnish all labor necessary and conduct a complete working test on the apparatus. The owner will furnish all fuel. Before the covering is applied the entire system must be tested for leaks to 20 pounds steam pressure, after which the regular working test at two pounds pressure must be made and the entire system must show a free flow of steam to, and of return water from, each radiator in the building.

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